Effects of Blending MTBE, Methanol, or Ethanol with Gasoline on Performance and Exhaust Emissions of SI Engines

by

Ali Mohammad Al-Dawood

A Thesis Presented to the

FACULTY OF THE COLLEGE OF GRADUATE STUDIES

KING FAHD UNIVERSITY OF PETROLEUM & MINERALS

DHAHRAN, SAUDI ARABIA

In Partial Fulfillment of the
Requirements for the Degree of

MASTER OF SCIENCE

In

MECHANICAL ENGINEERING

December, 1998

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DHAHRAN 31261, SAUDI ARABIA

COLLEGE OF GRADUATE STUDIES

This thesis, written by **Ali Mohammad Al-Dawood** under the direction of his Thesis Advisor and approved by his Thesis committee, has been presented to and accepted by the Dean of the College of Graduate Studies, in partial fulfillment of the requirements for the degree of **MASTER OF SCIENCE** in **MECHANICAL ENGINEERING.**

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Acknowledgement

Acknowledgement is due first to King Fahd University of Petroleum and Minerals for the support of this research.

I would like to express my deep appreciation to Dr. A. A. Al-Farayedhi who served as my major advisor. His continuous guidance was essential for completing this work successfully. I would like also to express my gratitude to Dr. J. Anabtawi of Saudi Aramco who served as a member of my thesis committee and whose assistance in obtaining and preparing the test fuels was very important. I also wish to thank the other members of my thesis committee Dr. M. A. Antar, and Dr. S. Ahmed. The thorough revision of the thesis and the comments by all the committee members were of high value to this work.

I also want to acknowledge the following companies that provided us with the test fuels and oxygenates: Saudi Aramco that provided the naphtha and reformate; the National Methanol Company (Ibn Sina) that provided the methanol and MTBE; and the Saudi Petrochemical Company (Sadaf) that provided the ethanol.

I extend my acknowledgement to KFUPM staff who helped in this research particularly M. K. Adham of the Heat Engines Laboratory for his assistance in setting up the test equipment, M. A. Ali of the Research Institute for performing the fuel hydrocarbon analysis, and A. S. Amer of the Oil Testing Center for testing the physical properties of fuels.

TABLE OF CONTENTS

LIST OF TABLES vi

LIST OF FIGURES vii

THESIS ABSTRACT xiii

NOMENCLATURE xv

CHAPTER 1: INTRODUCTION 1

1.1 BACKGROUND 1

1.1.1 Internal Combustion Engines 1

1.1.2 Important Engine-Related Definitions 2

1.1.3 Exhaust Emissions 4

1.1.4 Fuels and Additives 6

1.2 RESEARCH OBJECTIVES 7

CHAPTER 2: LITERATURE REVIEW 9

2.1 EFFECTS OF OPERATING VARIABLES ON ENGINE PERFORMANCE AND EMISSIONS 10

2.1.1 Effect of Equivalence Ratio 10

2.1.2 Effect of Spark Timing 12

2.1.3 Effects of Speed and Load 14

2.2 ROLE OF OXYGENATED FUEL PROPERTIES 17

2.2.1 Effect of Stoichiometric Air/Fuel Ratio 17

2.1.2 Effect of Heating Value 19

2.1.3 Effect of Latent Heat of Vaporization 19

2.1.4 Effect of Boiling Temperatures 20

2.1.5 Effect of Vapor Pressure 20

2.1.6 Effect of Flame Temperature and Burning Velocity 21

2.2 ENGINE DYNAMOMETER STUDIES 22

2.3 CHASSIS DYNAMOMETER STUDIES 26

CHAPTER 3: EXPERIMENTAL SET-UP 32

3.1 BACKGROUND 32

3.2 TEST FACILITIES 33

3.3 TEST FUELS 35

3.4 TEST CONDITIONS AND PROCEDURE 38

3.4.1 Spark Timing 38

3.4.2 Air/fuel Ratio 43

3.4.3 Other Operational Conditions 45

3.4.3 Test Procedure 46

CHAPTER 4: PERFORMANCE RESULTS AND DISCUSSION **48**

4.1 VARIABLE SPEED TEST 48

4.1.1 Maximum Engine Output 50

4.1.2 Brake Thermal Efficiency 53

4.2 VARIABLE EQUIVALENCE RATIO TEST 55

CHAPTER 5: EMISSIONS RESULTS AND DISCUSSION **83**

5.1 TESTED OPERATING VARIABLES 83

5.2 CO EMISSIONS 85

5.2.1 Effect of Equivalence Ratio on CO Emissions 85

5.2.2 Effect of Spark Timing on CO Emissions 86

5.2.3 Effects of Speed and Load on CO Emissions 87

5.2.4 CO Emissions at Idle Speed 88

5.3 HC EMISSIONS 89

5.3.1 Effect of Equivalence Ratio on HC Emissions 89

5.3.2 Effect of Spark Timing on HC Emissions 90

5.3.3 Effects of Speed and Load on HC Emissions 91

5.3.4 HC Emissions at Idle Speed 93

5.4 NOx EMISSIONS 93

5.4.1 Effect of Equivalence Ratio on NOx Emissions 94

5.4.2 Effect of Spark Timing on NOx Emissions 95

5.4.3 Effects of Speed and Load on NOx Emissions 96

5.4.4 NOx Emissions at Idle Speed 98

CHAPTER 6: CONCLUSIONS AND RECOMMENDATIONS **214**

6.1 SUMMARY OF PERFORMANCE RESULTS 214

6.1.1 Maximum Brake Mean Effective Pressure 214

6.1.2 Brake Thermal Efficiency at WOT 215

6.1.3 Brake Thermal Efficiency vs. Equivalence Ratio 215

6.2 SUMMARY OF EMISSIONS RESULTS 216

6.2.1 CO Emissions 216

6.2.2 HC Emissions 217

6.2.3 NOx Emissions 217

6.3 CONCLUSIONS 218

6.4 RECOMMENDATIONS 220

APPENDIXES **234**

REFERENCES **251**

List of Tables

Table	Title	Page
2-1:	Summary of the experimental research on the effects of MTBE, methanol, and ethanol on exhaust emissions.	31
3-1:	Properties of the oxygenates tested in this study.	38
3-2:	Properties of the test fuels.	39

List of Figures

Figure	Title	Page
1-1:	The basic structure of a spark-ignition engine.	2
2-1:	The typical influence of equivalence ratio on CO, HC, and NOx concentrations.	10
2-2:	Typical effect of equivalence ratio on maximum brake mean effective pressure (bmep) and brake specific fuel consumption (bsfc).	11
2-3:	The typical influence of spark timing on cylinder pressure at speed, throttle, and air/fuel ratio.	13
2-4:	Typical effect of spark timing on engine performance at speed, throttle position, and air/fuel ratio.	14
2-5:	Variation of brake mean effective pressure (bmep) and brake specific fuel consumption (bsfc) with engine speed at WOT conditions for a 3.8-liter six-cylinder spark-ignition engine.	15
2-6:	Typical variation of volumetric efficiency, fraction of input energy lost in heat transfer, and friction losses with engine speed.	16
2-7:	The influence of the equivalence ratio on the conversion efficiency of three-way catalysts.	18
2-8:	A schematic diagram illustrating the principle of closed-loop control of air/fuel ratio used with three-way catalysts.	18
2-9:	The adiabatic flame temperatures for alcohol, ethers, and major gasoline compounds.	22
3.1:	A schematic diagram illustrating the major testing facilities.	31
3-2:	The chemical structures of MTBE, methanol, and ethanol.	36
3-3:	Influence of ethanol content, aromatics content, and temperature on the water tolerance of ethanol/gasoline blends.	37
3-4:	D-86 distillation curves for the MTBE blends.	40
3-5:	D-86 distillation curves for the methanol blends.	41
3-6:	D-86 distillation curves for the ethanol blends.	42
3-7:	A schematic diagram for the ignition system of the test engine with the manual vacuum pump attached to the ignition controller.	44
3-8:	A schematic diagram for the fuel system of the test engine with the dc generator attached to the fuel-pressure actuator.	45
3-9:	An outline of the test procedure and test conditions.	47
4-1:	MBT spark timing at WOT for the MTBE blends.	58
4-2:	MBT spark timing at WOT for the methanol blends.	59

4-3:	MBT spark timing at WOT for the ethanol blends.	60
4-4:	Exhaust gas temperatures at WOT for the MTBE blends.	61
4-5:	Exhaust gas temperature at WOT for the methanol blends.	62
4-6:	Exhaust gas temperature at WOT for the ethanol blends.	63
4-7:	Brake torque and mean effective pressure at WOT for the MTBE blends.	64
4-8:	Brake torque and mean effective pressure at WOT for the methanol blends.	65
4-9:	Brake torque and mean effective pressure at WOT for the ethanol blends.	66
4-10:	The fluctuations in brake torque and engine noise level during a normal WOT variable-speed test.	67
4-11:	Brake torque and mean effective pressure at WOT for the best performing blends.	68
4-12:	Brake thermal efficiency at WOT for the MTBE blends.	69
4-13:	Brake thermal efficiency at WOT for the methanol blends.	70
4-14:	Brake thermal efficiency at WOT for the ethanol blends.	71
4-15:	Brake thermal efficiency at WOT for the best performing blends	72
4-16:	Effect of equivalence ratio on MBT timing for the MTBE blends.	73
4-17:	Effect of equivalence ratio on MBT timing for the methanol blends.	74
4-18:	Effect of equivalence ratio on MBT timing for the ethanol blends.	75
4-19:	Effect of equivalence ratio on exhaust gas temperature for the MTBE blends.	76
4-20:	Effect of equivalence ratio on exhaust gas temperature for the methanol blends.	77
4-21:	Effect of equivalence ratio on exhaust gas temperature for the ethanol blends.	78
4-22:	Effect of equivalence ratio on brake thermal efficiency for the MTBE blends.	79
4-23:	Effect of equivalence ratio on brake thermal efficiency for the methanol blends.	80
4-24:	Effect of equivalence ratio on brake thermal efficiency for the ethanol blends.	81
4-25:	Effect of equivalence ratio on thermal efficiency for the best performing blends.	82
5-1:	MBT timing vs. speed at a load of 80 Nm for the MTBE blends.	99
5-2:	MBT timing vs. speed at a load of 80 Nm for the methanol blends.	100
5-3:	MBT timing vs. speed at a load of 80 Nm for the ethanol blends.	101
5-4:	MBT timing vs. speed at a load of 120 Nm for the MTBE blends.	102
5-5:	MBT timing vs. speed at a load of 120 Nm for the methanol blends.	103
5-6:	MBT timing vs. speed at a load of 120 Nm for the ethanol blends.	104
5-7:	MBT timing vs. speed at a load of 160 Nm for the MTBE blends.	105
5-8:	MBT timing vs. speed at a load of 160 Nm for the methanol blends.	106
5-9:	MBT timing vs. speed at a load of 160 Nm for the ethanol blends.	107
5-10:	MBT timing vs. speed at a load of 200 Nm for the MTBE blends.	108

5-11:	MBT timing vs. speed at a load of 200 Nm for the methanol blends.	109
5-12:	MBT timing vs. speed at a load of 200 Nm for the ethanol blends.	110
5-13:	Effect of load on MBT timing at a speed of 2000 rpm for the MTBE blends.	111
5-14:	Effect of load on MBT timing at a speed of 2000 rpm for the methanol blends.	112
5-15:	Effect of load on MBT timing at a speed of 2000 rpm for the ethanol blends.	113
5-16:	Effect of equivalence ratio on CO emissions for the MTBE blends.	114
5-17:	Effect of equivalence ratio on CO emissions for the methanol blends.	115
5-18:	Effect of equivalence ratio on CO emissions for the ethanol blends.	116
5-19:	Effect of equivalence ratio on CO emissions for the least emitting blends.	117
5-20:	Effect of spark timing on exhaust gas temperature for the MTBE blends.	118
5-21:	Effect of spark timing on exhaust gas temperature for the methanol blends.	119
5-22:	Effect of spark timing on exhaust gas temperature for the ethanol blends.	120
5-23:	Effect of spark timing on CO emissions for the MTBE blends.	121
5-24:	Effect of spark timing on CO emissions for the methanol blends.	122
5-25:	Effect of spark timing on CO emissions for the ethanol blends.	123
5-26:	Effect of spark timing on CO emissions for the least emitting blends.	124
5-27:	Exhaust gas temperature vs. speed at a load of 160 Nm for the MTBE blends.	125
5-28:	Exhaust gas temperature vs. speed at a load of 160 Nm for the methanol blends.	126
5-29:	Exhaust gas temperature vs. speed at a load of 160 Nm for the ethanol blends.	127
5-30:	CO emissions vs. speed at a load of 80 Nm for the MTBE blends.	128
5-31:	CO emissions vs. speed at a Load of 120 Nm for the MTBE blends.	129
5-32:	CO emissions vs. speed at a load of 160 Nm for the MTBE blends.	130
5-33:	CO emissions vs. speed at a load of 200 Nm for the MTBE blends.	131
5-34:	Effect of load on CO emissions at a speed of 2000 rpm for the MTBE blends.	132
5-35:	CO emissions vs. speed at a load of 80 Nm for the methanol blends.	133
5-36:	CO emissions vs. speed at a Load of 120 Nm for the methanol blends.	134
5-37:	CO emissions vs. speed at a load of 160 Nm for the methanol blends.	135
5-38:	CO emissions vs. speed at a load of 200 Nm for the methanol blends.	136
5-39:	Effect of load on CO emissions at a speed of 2000 rpm for the methanol blends.	137
5-40:	CO emissions vs. speed at a load of 80 Nm for the ethanol blends.	138
5-41:	CO emissions vs. speed at a Load of 120 Nm for the ethanol blends.	139
5-42:	CO emissions vs. speed at a load of 160 Nm for the ethanol blends.	140
5-43:	CO emissions vs. speed at a load of 200 Nm for the ethanol blends.	141

5-44:	Effect of load on CO emissions at a speed of 2000 rpm for the ethanol blends.	142
5-45:	CO emissions vs. speed at a load of 80 Nm for the least emitting.	143
5-46:	CO emissions vs. speed at a Load of 120 Nm for the least emitting blends.	144
5-47:	CO emissions vs. speed at a load of 160 Nm for the least emitting blends.	145
5-48:	CO emissions vs. speed at a load of 200 Nm for the least emitting blends.	146
5-49:	Exhaust temperatures at idle speed as a function of spark timing for selected blends.	147
5-50:	Effect of spark timing on CO emissions at idle speed for the MTBE blends.	148
5-51:	Effect of spark timing on CO emissions at idle speed for the methanol blends.	149
5-52:	Effect of spark timing on CO emissions at idle speed for the ethanol blends.	150
5-53:	Effect of spark timing on CO emissions at idle speed for the least emitting blends.	151
5-54:	Effect of equivalence ratio on HC emissions for the MTBE blends.	152
5-55:	Effect of equivalence ratio on HC emissions for the methanol blends.	153
5-56:	Effect of equivalence ratio on HC emissions for the ethanol blends.	154
5-57:	Effect of equivalence ratio on HC emissions for the least emitting blends.	155
5-58:	Effect of spark timing on HC emissions for the MTBE blends.	156
5-59:	Effect of spark timing on HC emissions for the methanol blends.	157
5-60:	Effect of spark timing on HC emissions for the ethanol blends.	158
5-61:	Effect of spark timing on HC emissions for the least emitting blends.	159
5-62:	Exhaust HC emissions vs. speed at a load of 80 Nm for the MTBE blends.	160
5-63:	Exhaust HC emissions vs. speed at a load of 120 Nm for the MTBE blends.	161
5-64:	Exhaust HC emissions vs. speed at a load of 160 Nm for the MTBE blends.	162
5-65:	Exhaust HC emissions vs. speed at a load of 200 Nm for the MTBE blends.	163
5-66:	Exhaust HC emissions vs. speed at a load of 80 Nm for the methanol blends.	164
5-67:	Exhaust HC emissions vs. speed at a load of 120 Nm for the methanol blends.	165
5-68:	Exhaust HC emissions vs. speed at a load of 160 Nm for the methanol blends.	166
5-69:	Exhaust HC emissions vs. speed at a load of 200 Nm for the methanol blends.	167
5-70:	Exhaust HC emissions vs. speed at a load of 80 Nm for the ethanol blends.	168
5-71:	Exhaust HC emissions vs. speed at a load of 120 Nm for the ethanol blends.	169
5-72:	Exhaust HC emissions vs. speed at a load of 160 Nm for the ethanol blends.	170
5-73:	HC emissions vs. speed at a load of 200 Nm for the ethanol blends.	171
5-74:	HC emissions vs. speed at a load of 80 Nm for the least emitting blends.	172
5-75:	HC emissions vs. speed at a load of 120 Nm for the least emitting.	173
5-76:	HC emissions vs. speed at a load of 160 Nm for the least emitting blends.	174

5-77:	HC emissions vs. speed at a load of 200 Nm for the least emitting blends.	175
5-78:	Effect of load on HC emissions at a speed of 2000 rpm for the MTBE blends.	176
5-79:	Effect of load on HC emissions at a speed of 2000 rpm for the methanol blends.	177
5-80:	Effect of load on HC emissions at a speed of 2000 rpm for the ethanol blends.	178
5-81:	Effect of load on HC emissions at a speed of 2000 rpm for the least emitting blends.	179
5-82:	Effect of spark timing on HC emissions at idle speed for the MTBE blends.	180
5-83:	Effect of spark timing on HC emissions at idle speed for the methanol blends.	181
5-84:	Effect of spark timing on HC emissions at idle speed for the ethanol blends.	182
5-85:	Effect of spark timing on HC emissions at idle speed for the least emitting blends.	183
5-86:	Effect of equivalence ratio on NOx emissions for the MTBE blends.	184
5-87:	Effect of equivalence ratio on NOx emissions for the methanol blends.	185
5-88:	Effect of equivalence ratio on NOx emissions for the ethanol blends.	186
5-89:	Effect of equivalence ratio on NOx emissions for the least emitting blends.	187
5-90:	Effect of spark timing on NOx emissions for the MTBE blends.	188
5-91:	Effect of spark timing on NOx emissions for the methanol blends.	189
5-92:	Effect of spark timing on NOx emissions for the ethanol blends.	190
5-93:	Effect of spark timing on NOx emissions for the least emitting blends.	191
5-94:	NOx emissions vs. speed at a load of 80 Nm for the MTBE blends.	192
5-95:	NOx emissions vs. speed at a load of 120 Nm for the MTBE blends.	193
5-96:	NOx emissions vs. speed at a load of 160 Nm for the MTBE blends.	194
5-97:	NOx emissions vs. speed at a load of 200 Nm for the MTBE blends.	195
5-98:	Effect of load on NOx emissions at a speed of 2000 rpm for the MTBE blends.	196
5-99:	NOx emissions vs. speed at a load of 80 Nm for the methanol blends.	197
5-100:	NOx emissions vs. speed at a load of 120 Nm for the methanol blends.	198
5-101:	NOx emissions vs. speed at a load of 160 Nm for the methanol blends.	199
5-102:	NOx emissions vs. speed at a load of 200 Nm for the methanol blends.	200
5-103:	Effect of load on NOx emissions at a speed of 2000 rpm for the methanol blends.	201
5-104:	NOx emissions vs. speed at a load of 80 Nm for the ethanol blends.	202
5-105:	NOx emissions vs. speed at a load of 120 Nm for the ethanol blends.	203
5-106:	NOx emissions vs. speed at a load of 160 Nm for the ethanol blends.	204
5-107:	NOx emissions vs. speed at a load of 200 Nm for the ethanol blends.	205
5-108:	Effect of load on NOx emissions at a speed of 2000 rpm for the ethanol blends.	206
5-109:	NOx emissions vs. speed at a load of 80 Nm for the least emitting blends.	207

5-110:	NOx emissions vs. speed at a load of 120 Nm for the least emitting blends.	208
5-111:	NOx emissions vs. speed at a load of 160 Nm for the least emitting blends.	209
5-112:	NOx emissions vs. speed at a load of 200 Nm for the least emitting blends.	210
5-113:	Effect of spark timing on NOx emissions at idle speed for the MTBE blends.	211
5-114:	Effect of spark timing on NOx emissions at idle speed for the methanol blends.	212
5-115:	Effect of spark timing on NOx emissions at idle speed for the ethanol blends.	213
6-1:	Effect of the oxygenates on brake mean effective pressure at 2500 rpm and WOT.	221
6-2:	Effect of the oxygenates on brake thermal efficiency at 2500 rpm and WOT.	222
6-3:	Effect of the oxygenates on brake thermal efficiency for stoichiometric mixtures (Φ=1.0) at 2000 rpm.	223
6-4:	Effect of the oxygenates on CO emissions at low load (340 kPa) and mid-range speed (2000 rpm).	224
6-5:	Effect of the oxygenates on CO emissions at high load (680 kPa) and mid-range speed (2000 rpm).	225
6-6:	Effect of the oxygenates on CO emissions at idle speed (1000 rpm) with spark timing at 3 degrees btc.	226
6-7:	Effect of the oxygenates on CO emissions at idle speed (1000 rpm) with spark timing at 9 degrees btc.	227
6-8:	Effect of the oxygenates on HC emissions at mid-range speed (2000 rpm) and low load (340 kPa).	228
6-9:	Effect of the oxygenates on HC emissions at mid-range speed (2000 rpm) and high load (680 kPa).	229
6-10:	Effect of the oxygenates on HC emissions at idle speed (1000 rpm) with spark timing at 3 degrees btc.	230
6-11:	Effect of the oxygenates on HC emissions at idle speed (1000 rpm) with spark timing at 9 degrees btc.	231
6-12:	Effect of the oxygenates on NOx emissions at mid-range speed (2000 rpm) and low load (340 kPa).	232
6-13:	Effect of the oxygenates on NOx emissions at mid-range speed (2000 rpm) and high load (680 kPa).	233

Thesis Abstract

Name of Student	:	Ali Mohammad Al-Dawood
Title of Study	:	Effects of Blending MTBE, Methanol, or Ethanol with Gasoline on Performance and Exhaust Emissions of SI Engines
Major Field	:	Mechanical Engineering
Date of Degree	:	December 30, 1998

The current experimental study aims to examine the effects of using oxygenates as a replacement of lead additives in gasoline on performance and exhaust emissions of a typical SI engine. The tested oxygenates are MTBE, methanol, and ethanol. These oxygenates were blended with a base unleaded fuel in three ratios (10, 15, and 20 vol.%). The engine maximum output, thermal efficiency, CO emissions, HC emissions, and NOx emissions were all evaluated at a variety of engine operating conditions using an engine dynamometer set-up. The results of the oxygenated blends were compared to those of the base fuel and of a leaded fuel prepared by adding TEL to the base.

When compared to the base and leaded fuels, the oxygenated blends improved the engine brake thermal efficiency. The leaded fuel performed better than the oxygenated blends in terms of the maximum output of the engine except in the case of 20 vol.% methanol and 15 vol.% ethanol blends. With respect to the base fuel, the addition of oxygenates decreased the CO emissions, decreased the HC emissions at most operating conditions, but generally increased the NOx emissions except for 10 vol.% ethanol blend. The emissions results for the leaded fuel were comparable to those of the base.

The methanol blends performed better than the other oxygenated blends in terms of engine output and thermal efficiency. The MTBE blends were the best performers among the oxygenated blends in terms of HC emissions while the ethanol blends were the best performers in terms of CO and NOx emissions.

MASTER OF SCIENCE DEGREE

KING FAHD UNIVERSITY OF PETROLEUM AND MINERALS
Dhahran, Saudi Arabia

Date
December 30, 1998

خلاصة الرسالة

اسم الطالب : علي محمد أحمد الداوود

عنوان الدراسة : تأثير إضافة بيوتال الأثير الثلاثي الميثيلي أو الميثانول أو الإيثانول مع البنزين على الأداء ونواتج الاحتراق لمحركات الإشعال بالشرارة.

التخصص : هندسة ميكانيكية

تاريخ الشهادة : ١٩٩٨/١٢/٣٠م

يهدف هذا البحث التجريبي إلى اختبار تأثير استخدام بعض المواد المحتوية على أكسجين (Oxygenates) كبديل لمركبات الرصاص في وقود محركات الاحتراق الداخلي ذات الإشعال بالشرارة. المواد التي تم اختبارها في هذا البحث هي بيوتال الأثير الثلاثي الميثيلي (MTBE) و الميثانول و الإيثانول. وقد تم مزج هذه المواد مع وقود مرجعي لا يحتوي على الرصاص بثلاث نسب مختلفة (١٠، ١٥، و ٢٠ % حجمياً). تم قياس مردود المحرك الأقصى و كفاءته الحرارية ونسبة كلٍ من أول أكسيد الكربون والهيدروكربونات وأكاسيد النيتروجين في نواتج الاحتراق وذلك في ظروف تشغيل متنوعة على داينمومتر خاص باختبار المحركات. كما تمت مقارنة نتائج المزائج المختلفة بتلك الخاصة بالوقود المرجعي والوقود المحتوي على الرصاص والذي تم إعداده بإضافة رابع ايثيل الرصاص (TEL) الى الوقود المرجعي.

بالمقارنة مع الوقود المرجعي والوقود المحتوي على الرصاص فإن المزائج المحتوية على أكسجين أدت إلى تحسين الكفاءة الحرارية بشكل ملحوظ. غير أن المردود الأقصى للمحرك إجمالا كان أعلى في حالة الوقود المحتوي على رصاص منه في حالة المزائج المحتوية على أكسجين. ومقارنةً مع الوقود المرجعي فإن المزائج المحتوية على أكسجين أدت إلى تقليل نسبة أول أكسيد الكربون في نواتج الاحتراق وتقليل نسبة الهيدروكربونات أيضاً في معظم ظروف التشغيل غير أنها في المقابل زادت من أكاسيد النيتروجين إلا في حالة المزيج المحتوي على نسبة منخفضة من الإيثانول. وبالنسبة لنسب هذه الملوثات في حالة الوقود المحتوي على رصاص فإنها قريبة من تلك الخاصة بالوقود المرجعي.

أظهرت النتائج أن أداء مزائج الميثانول كان أفضل من باقي المزائج في ما يتعلق بمردود المحرك الأقصى والكفاية الحرارية. وبشكل عام فقد تميزت مزائج بيوتال الأثير الثلاثي الميثيلي بأقل نسبة هيدروكربونات في نواتج الاحتراق بينما تميزت مزائج الإيثانول بأقل نسبة لأول أكسيد الكربون وأكاسيد النيتروجين.

درجة الماجستير في العلوم

جامعة الملك فهد للبترول والمعادن

الظهران ، المملكة العربية السعودية

التاريخ

١٩٩٨/١٢/٣٠ م

Nomenclature

Φ	Equivalence ratio
bc	Bottom center
bmep	Brake mean effective pressure
bsfc	Brake specific fuel consumption
btc	Before top center
CI	Compression ignition
CL	Chemiluminescent analyzer
CO	Carbon monoxide
ETBE	Ethyl tertiary butyl ether
FID	Flame ionization detector
FTP	Federal test procedure
HC	Hydrocarbons
MBT	Maximum brake torque spark timing
MTBE	Methyl tertiary butyl ether
NDIR	Nondispersive infrared analyzer
NOx	Oxides of nitrogen
Rvp	Reid vapor pressure
SI	Spark ignition
tc	Top center
TEL	Tetra-ethyl lead
TML	Tetra-methyl lead
TWC	Three-way catalyst
WOT	Wide-open throttle

Chapter 1

Introduction

1.1 Background

1.1.1 Internal Combustion Engines

The internal combustion engines are the driving force in most today's automotive applications. In these engines, the combustion of a mixture of air and fuel takes place in a confined area called the combustion chamber. Heat energy is released as a result of the oxidation of fuel molecules during the combustion process. The released heat energy causes the combustion gases to expand forcing the piston downward and thus exerting a rotational force on the crankshaft of the engine.

The process of converting the fuel energy into mechanical work through combustion takes place in very fast repeating cycles. The cycle usually involves five processes: the induction, the compression, the combustion, the expansion, and the exhaust. In the four-stroke engine, the cycle is performed in four piston movements (two upward and two downward), called strokes, completed in two crankshaft revolutions. In the first stroke, the piston moves from the uppermost position, called the top center (tc), to the lowermost position, called the bottom center (bc), inducting the air/fuel mixture (or the air only depending on the type of the engine) through the open intake valve. In the second stroke, the piston moves upward compressing the mixture (or air only) while the valves are closed. Before the end of the compression stroke, the combustion process starts and continues well in the next stroke where the heat release from combustion expands the gases and forces the piston downward. In the last stroke, the combustion products are forced outside the cylinder through the open exhaust valve by the upward moving piston.

Since only one stroke of the cycle produces power, a smooth rotation of the crankshaft requires the engine to be built with several cylinders performing the cycle processes at different intervals. The commencement of the combustion process is triggered either by an external spark as in the case of spark-ignition (SI) engines or by the injection of the fuel into a highly compressed air as in the case of compression-ignition (CI) engines. Figure 1-1 shows the basic structure of a spark-ignition engine [1, 2, 3, 4].

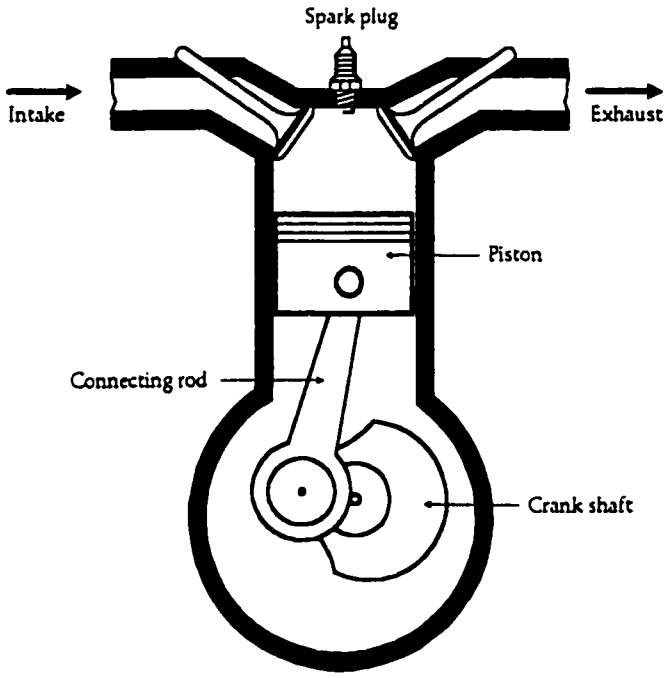

Figure 1-1: The basic structure of a spark-ignition engine.

1.1.2 Important Engine-Related Definitions

Because of incomplete combustion, not all the energy of the fuel supplied to the engine is released. Furthermore, not all the heat energy released during combustion is converted into mechanical work. About two thirds of this heat is lost through the cylinder walls and during the exhaust process. Part of the work produced during the expansion process

is consumed by the engine itself during the induction, the compression, and the exhaust processes and to overcome the mechanical friction between the engine moving components. The effectiveness of the engine in converting the fuel energy into mechanical work is measured by the brake thermal efficiency η_{bt} defined as the ratio between the engine brake power and the rate of fuel energy input.

$$\eta_{bt} = \frac{P_{brake}(kW)}{\dot{m}_f(kg/s)\, Q_{LHV}(kJ/kg)} \qquad \text{(Equation 1-1)}$$

where P_{brake} is the engine brake power, $\dot{m}_f$ is the fuel consumption rate, and Q_{LHV} is the lower heating value of the fuel.

The optimization of the brake thermal efficiency involves optimizing the induction, the compression, the combustion, and the exhaust processes along with minimizing the friction losses.

Another important parameter used as an indicator to the efficiency of engines is the specific fuel consumption defined as the ratio between the fuel consumption rate and the engine brake power. Contrary to the definition of brake thermal efficiency, the brake specific fuel consumption (bsfc) has a unit (usually g/kWh) and is valid for comparison only if the fuels used have the same energy content. In case of different fuels having different energy content, it is imperative to use the definition of the brake thermal efficiency instead of the bsfc.

$$bsfc(g/kWh) = \frac{\dot{m}_f(g/h)}{P_{brake}(kW)} = \frac{3600}{\eta_{bt}\, Q_{LHV}(MJ/kg)} \qquad \text{(Equation 1-2)}$$

The engine output is usually measured in terms of the maximum torque exerted on the crankshaft of the engine. The torque is also used as a measure of the load applied to the engine. More useful parameter, which is independent on the engine size, is usually used instead of the torque. This parameter is called the brake mean effective pressure (bmep) and calculated by dividing the work per cycle by the engine displacement volume. For

four-stroke engine, the brake mean effective pressure can be calculated by the following relation:

$$\text{bmep(kPa)} = \frac{P_{brake}(\text{kW}) \times 2000}{V_d(\text{Liter})\, N(\text{rev/s})} \qquad \text{(Equation 1-3)}$$

where V_d is the engine displacement volume and N is the engine rotational speed.

The engine maximum torque is considerably affected by the effectiveness of the induction process. In naturally aspirated engines, the mass of the charge drawn into the cylinder is usually less than the mass of an equivalent volume of air that fills the displaced volume of the cylinder. The ratio between the two masses is called the volumetric efficiency and is usually between 75 to 90 percent. This deficiency is mainly a result of the flow restrictions in the induction line, and varies with varying operating conditions [1, 2, 3, 4].

1.1.3 Exhaust Emissions

The complete combustion of hydrocarbon fuels during the combustion process in internal combustion engines can not be achieved. Unburned hydrocarbons (HC) are emitted with the exhaust gases due to this incomplete combustion. Improper mixing and preparation of air and fuel, flame quenching, and chamber deposits are some of the reasons behind the emission of unburned hydrocarbons. Hydrocarbons cause irritation to eyes and contain some compounds believed to cause cancer (i.e. carcinogenic). Hydrocarbons are classified either as reactive or non-reactive based on their role in the chemical reaction that lead to the production of the photochemical smog as will be discussed below. Saturated hydrocarbons such as paraffins (C_nH_{2n+2}, single-bond open-chain structure) and naphthenes (C_nH_{2n}, single-bond ring structure) are considered non-reactive, while unsaturated hydrocarbons, particularly olefins (C_nH_{2n}, double-bond open-chain structure) and to lesser degree aromatics (C_nH_{2n-6}, benzene-ring structure), are considered reactive hydrocarbons [1, 2, 3, 4, 6].

The formation of nitrogen oxides (NOx) during the combustion process is a result of the extremely high temperatures existing inside the combustion flame. In spark ignition engines, the major part of these oxides is the nitric oxide (NO) beside a small percentage (about 2%) of nitrogen dioxide (NO_2). In the presence of sunlight, nitrogen dioxide (NO_2) breaks into nitric oxide and an oxygen atom (O) that combines with atmospheric oxygen (O_2) to form the ozone (O_3). The presence of unsaturated hydrocarbons, such as olefins and aromatics, inhibits the recombination of ozone (O_3) with nitric oxide (NO) to form the nitrogen dioxide (NO_2) again. Instead, the ozone reacts with these reactive hydrocarbons producing certain chemicals such as the acrolein and peroxyacetyl nitrate (PAN). These chemicals along with others involved in the reactions are collectively known as the photochemical smog. The photochemical smog is usually experienced on sunny mornings when the atmospheric concentration of nitrogen dioxide (NO_2) is high. It causes the eye irritation and a reduction in visibility. Nitrogen oxides play a role also in the formation of acidic rain through the reaction of nitrogen dioxide (NO_2) with the atmospheric water vapor that produces the nitric acid (HNO_3) [1, 6].

Carbon monoxide (CO) is a product of incomplete oxidation of the fuel. The formation of carbon monoxide is highly dependent on the availability of oxygen at the time of combustion. Carbon monoxide is a colorless odorless toxic gas that, when enters the blood stream, hinders the transport of oxygen from the lungs to the rest of the body [2, 6].

Different measures have been taken to control these emissions. Several in-combustion and after-combustion emission control techniques are currently in use. In spark-ignition engines, proper mixing and preparation of air and fuel charge have been attained through the use of gasoline injection and the closed-loop electronic control systems. New concepts related to the design of combustion environment, like the stratified-charge engine, have been introduced. Exhaust gas re-circulation has been used to control the combustion temperature and therefore controlling the formation of oxides of nitrogen.

After-combustion measures include air injection in the exhaust manifold and using catalytic converters. Air injection promotes the oxidation of incompletely burned species, and catalytic converters treat the harmful emissions and convert them to non-harmful species [2].

1.1.4 Fuels and Additives

The gasoline fuel used in spark-ignition internal combustion engines is a blend of many hydrocarbon compounds derived from distillation of crude oil. The main constituents of typical gasoline are the paraffins (50-80 vol. %), olefins (0-15 vol. %), and aromatics (15-40 vol. %). Several additives are blended with gasoline to enhance certain properties necessary for satisfactory performance. Properties like distillation characteristics, vapor pressure, anti-corrosion, and most importantly the anti-knock quality can be substantially improved by certain additives. The anti-knock quality is the limiting factor for the engine compression ratio on which the maximum theoretical engine efficiency is dependent. Knock is a high-pitch metallic noise that results from spontaneous ignition of unburned mixture before it is approached by the advancing flame during the combustion process. Knock may result in severely damaging effects on engine parts and performance. The anti-knock quality of the fuel can be enhanced by the addition of lead alkyls, either tetra-ethyl lead (TEL) or tetra-methyl lead (TML), in small quantities usually regulated by local legislators in each country. The lead additives significantly enhance the antiknock quality of gasoline, but on the other hand, result in the formation and emission of toxic lead compounds. Another disadvantage of using lead additives is their damaging effect on the active materials of the catalytic devices used to control emissions [2]. For these reasons, unleaded or reduced-lead fuels are currently required in many countries around the world.

An alternative to the use of lead additives is the reforming of lower-octane (i.e. lower antiknock quality) fuel compounds, such as naphthenes and normal-paraffins, into higher-octane compounds such as iso-paraffins and aromatics. The reforming process is

very effective in enhancing the antiknock quality of the fuel, however, it involves a very large investment in reforming plants and processes. Fuels containing high proportions of aromatics and olefins produce relatively higher concentrations of hydrocarbon compounds that have a potential to participate in reactions leading to the production of the harmful photochemical smog. In addition, some aromatic compounds are known to be carcinogenic and nerve toxins. For these reasons, the current trend favors the lowering of aromatics content in gasoline [7]. Aromatics have good antiknock properties, therefore, reducing their content lowers the overall antiknock quality of the fuel [8].

A more recent practice is to enhance the anti-knock property of the fuel by using certain high-octane oxygen-containing organic compounds called oxygenates. The most commonly used oxygenates are MTBE (methyl tertiary butyl ether, C_4H_9-O-CH_3), methanol (CH_3-OH), and ethanol (C_2H_5-OH). MTBE is manufactured from isobutane ($(CH_3)_3CH$) and methanol, while methanol is manufactured from natural gas or synthesized from a variety of materials such as coal, municipal wastes, and biomass. Ethanol is derived from the direct fermentation of sugars, fermentation of starches and cellulose after chemical or enzymatic pretreatment, or made from petroleum sources [9].

These three oxygenates have different chemical and physical properties when compared to gasoline. These differences are expected to influence the performance and combustion products of gasoline-oxygenate blends.

1.2 Research Objectives

The lead additives to gasoline are no longer used in many countries around the world. Many other countries are now phasing-down the lead in gasoline. Although the leaded fuel is still in use in Saudi Arabia, several plans are considered to phase-down the lead. The use of oxygenates to replace the lead additives in gasoline is considered now as an option in Saudi Arabian refineries [10]. This current experimental study is aimed to help in understanding the effect of the most popular oxygenates on performance and exhaust

emissions of typical automotive engines. The study offers a comparison between the oxygenated and leaded fuels in terms of engine performance and emissions.

Oxygenating gasoline with ethers or alcohol has been investigated widely. However, as indicated in the literature review, there is a remarkable variability in the results particularly those for exhaust emissions. This variability makes the evaluation of the actual effect of oxygenates very difficult. One important source of this variability in results is the diversity of test methods and conditions. Most researchers studied the effects of oxygenates on engine performance and emissions without applying any modification to the engine. In other words, the engines were not adjusted to work with the new fuels that have different properties. Furthermore, most studies were performed at very limited range of operating conditions despite the very broad range of conditions in which the engine normally operates. These are some of the factors that have contributed to the variability and, in several cases, conflict in reported results.

In the current study, the engine performance parameters and exhaust emissions were studied extensively at very broad range of operating conditions. The effects of the important operating variables were evaluated within the practical operating range. All the experiments were well-controlled and modifications were applied as necessary for each tested fuel. The experiments were designed to give an objective basis for comparison between leaded and oxygenated fuels, and between the different oxygenates.

Chapter 2

Literature Review

The use of oxygenates has drawn a considerable attention of researchers both in universities and industry because of its potential impact on environment and fuel economics. Extensive research has been carried out in the last two decades to study the influence of blending ethers and alcohol with gasoline on fuel properties and emissions from internal combustion engines. Researchers studied the potential of some alcoholic compounds, specially methanol and ethanol, as alternative fuels or as gasoline blending components [22, 23, 25, 26, 27, 28, 30, 34, 35, 36]. Researchers also studied the influence of adding ethers specially methyl tertiary butyl ether (MTBE) to gasoline as an octane extender instead of lead alkyls [24, 28, 30, 31, 32, 33]. Some research has been also devoted to studying the impact of oxygenates (ether or alcohol) on engine performance parameters such as maximum output, thermal efficiency, and/or fuel consumption [22, 23, 24, 27].

In this literature review, the effects of the engine operating variables on the performance and exhaust emissions will be discussed first. This discussion will help in explaining the trends of the experimental results. The effects of oxygenates on fuel properties and how these effects can influence the formation of emissions and engine performance will also be discussed. This discussion will help in understanding the differences between the experimental results for the various tested fuels and will form the basis on which these differences are explained. Several experimental studies, including studies conducted on engines and studies conducted on vehicles, will be reviewed. The review will emphasize the test methodology, equipment, and overall results along with any given explanations. The results of these studies are quantitatively summarized as possible in Table 2-1 at the end of this chapter.

2.1 Effects of Operating Variables on Engine Performance and Emissions

2.1.1 Effect of Equivalence Ratio

The air/fuel ratio is the mass ratio between air and fuel in the mixture supplied to the engine. The stoichiometric air/fuel ratio defines the minimum amount of air required to burn the fuel completely. The ratio between the stoichiometric air/fuel ratio and the actual air/fuel ratio is called the equivalence ratio (Φ). The equivalence ratio has a significant influence on the formation of exhaust emissions.

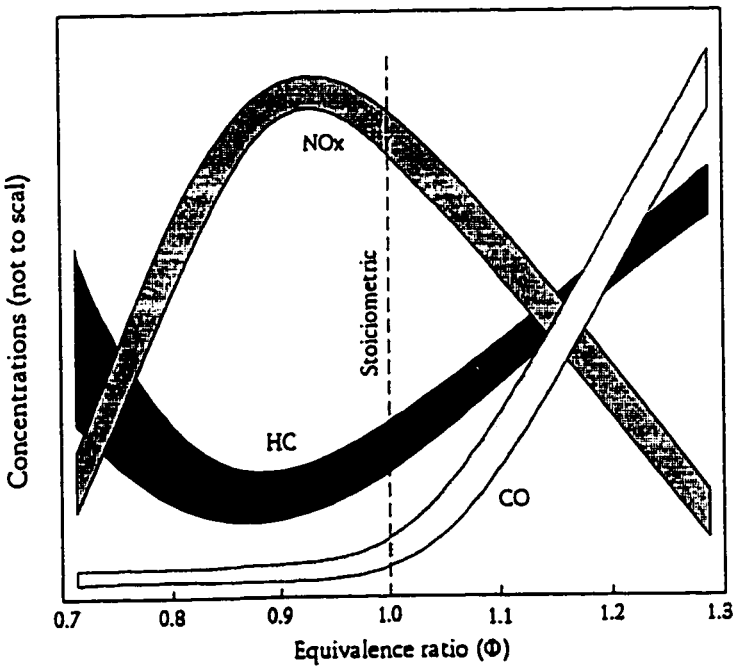

Figure 2-1: The typical influence of equivalence ratio on CO, HC, and NOx concentrations [3].

Figure 2-1 shows qualitatively the general trends of formation of NOx, HC, and CO emissions as a function of equivalence ratio (Φ). At the lean side just after the stoichiometric mixture, the figure shows a change of trend for NOx and HC concentrations, while CO continues to decrease slightly. The concentration of nitrogen oxides (NOx) is mainly influenced by the flame temperature and availability of oxygen.

The combined effect of these two is greatest at about Φ=0.9 where NOx concentration becomes maximum. Hydrocarbon emissions (HC) are influenced mainly by the combustion quality and availability of oxygen. On the rich side (Φ>1), the fuel has no enough oxygen to fully oxidize which results in more emitted unburned hydrocarbons. On the very lean side (Φ<0.85), the combustion quality deteriorates and misfire takes place resulting in more emitted unburned hydrocarbons. The best balance between combustion quality and availability of oxygen exists at about Φ=0.9 where HC concentration becomes minimum. Carbon monoxide (CO) is mainly influenced by the availability of oxygen during combustion, therefore, CO is expected to continue to decrease as the mixture becomes leaner [2, 4, 5, 11].

The equivalence ratio has also an appreciable effect on the engine performance. Figure 2-2 shows the typical effect of equivalence ratio on the maximum output and specific fuel consumption of a spark ignition engine. Maximum engine output is attained at the rich side of equivalence ratio as a result of the higher heat release rate. On the other

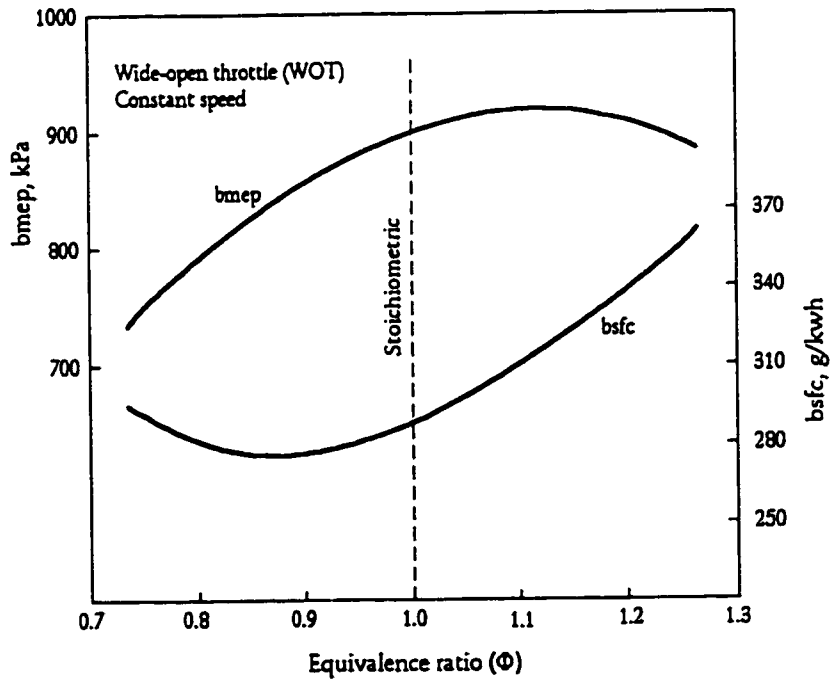

Figure 2-2: Typical effect of equivalence ratio on maximum brake mean effective pressure (bmep) and brake specific fuel consumption (bsfc) [4].

hand, the least specific fuel consumption is attained at the lean side due to the better combustion of fuel molecules which results in higher thermal efficiency.

2.1.2 Effect of Spark Timing

The spark timing is the crankshaft angle at which the ignition spark is produced between the electrodes of the spark plugs. The spark usually occurs before top center near the end of compression stroke. The spark timing has a considerable influence on the combustion characteristics and therefore on the engine performance and combustion products. Figure 2-3 shows the variation of cylinder pressure with three different spark timings. The too advanced spark timing (50 degrees btc in this case) causes the cylinder pressure to increase substantially and rapidly before the end of the compression stroke (i. e. before top center). This increases the work lost in the compression process, and therefore, decreases the net useful engine work. On the other hand, the too retarded spark timing (10 degrees btc) results in a lower peak pressure occurring very late in the expansion process. This reduces the work transfer from the expanding gases to the piston. The optimum spark timing (30 degrees btc in this case) gives a satisfactorily high cylinder pressure with its peak occurring just after the top center (beginning of expansion stroke). This insures a minimum compression work and maximum work transfer during the expansion stroke. These variations would have a significant impact on the engine performance parameters [2, 3, 4].

Figure 2-4 shows a typical effect of spark timing on the engine maximum output and thermal efficiency. The engine performance is improved as the spark timing is advanced until reaching certain point after which the performance starts to decline. The decline in engine performance is a result of the increasing pressure on the engine pistons during the compression stroke. As the spark timing is advanced, the cylinder pressure and temperature increase substantially causing the end gas to auto-ignite before it is consumed by the advancing flame. This auto-ignition leads to the production of damaging audible sound waves. These undesirable sound waves are called knock. The

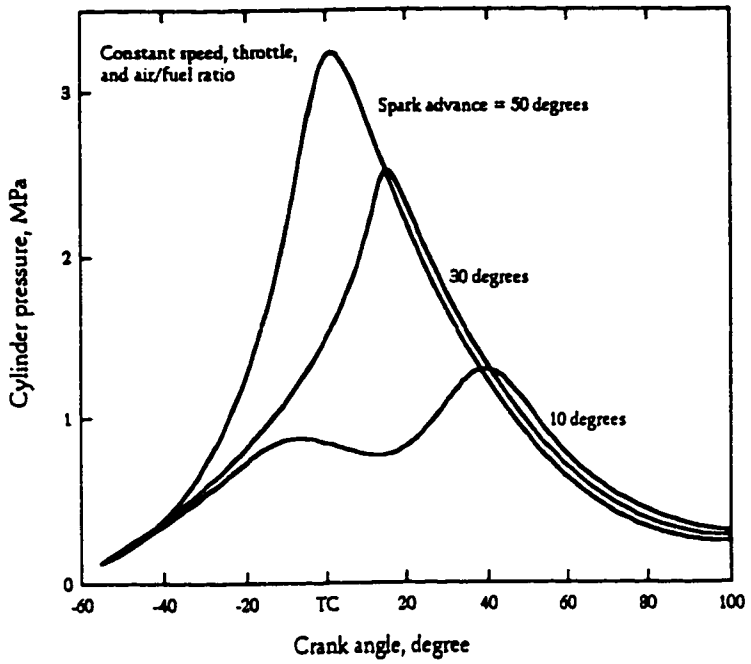

Figure 2-3: The typical influence of spark timing on cylinder pressure at constant speed, throttle, and air/fuel ratio [2].

spark timing at which the engine performance is optimum, and at which the knock is not encountered, is called the maximum brake torque (MBT) timing. Along with the attained maximum brake torque, the MBT timing is characterized by the minimum specific fuel consumption and therefore maximum brake thermal efficiency [2, 4].

The variations in cylinder pressure due to the spark timing will affect the formation of exhaust emissions because of their strong dependence on combustion conditions. As the spark timing is advanced towards the MBT timing, the peak cylinder pressure increases and moves closer to the optimum angle resulting in higher combustion temperature and more work transfer during expansion stroke. At higher combustion temperatures, more nitrogen oxides (NOx) are formed. Furthermore, the increase in work transfer to the pistons results in higher thermal efficiency and lower exhaust temperatures. Exhaust unburned hydrocarbons (HC) continues to oxidize throughout the expansion and exhaust processes. The oxidation reactions of unburned hydrocarbons favor higher exhaust temperatures, and therefore, decreasing the exhaust temperature decreases the

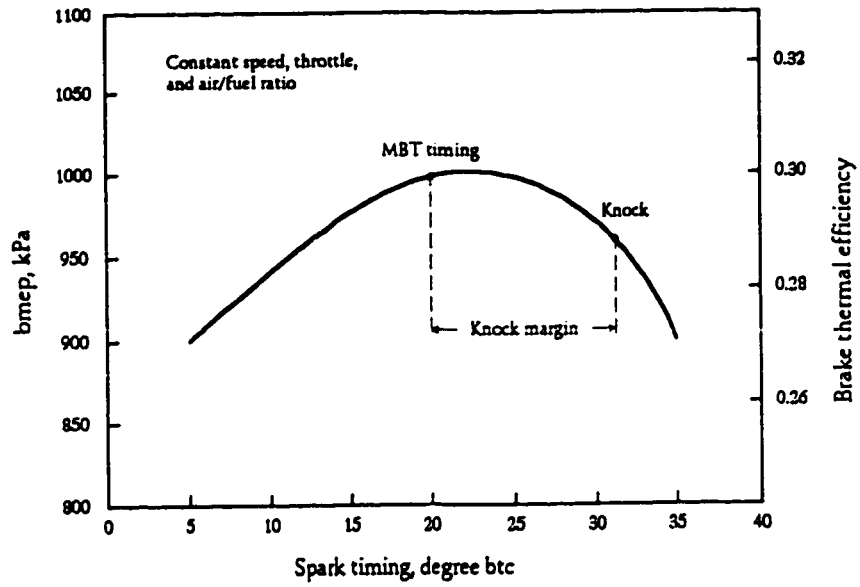

Figure 2-4: Typical effect of spark timing on engine performance at constant speed, throttle position, and air/fuel ratio [4].

amount of hydrocarbons oxidized. The effect of exhaust temperature on carbon monoxide (CO) is similar to hydrocarbons, but carbon monoxide requires higher exhaust temperatures and longer time (residence) to oxidize [2, 11].

2.1.3 Effects of Speed and Load

Engines normally operate on a very wide range of speed and load to satisfy the variable driving requirements. The engine performance characteristics vary considerably with engine speed. Figure 2-5 shows the typical variation of brake mean effective pressure, which represents the engine output, and brake specific fuel consumption with engine speed at wide-open throttle. The maximum output and minimum specific fuel consumption are obtained at mid-range speeds where the volumetric efficiency is maximum and the net power losses are minimum. Figure 2-6 shows the typical variation of volumetric efficiency, heat losses to the cylinder walls, and mechanical friction losses with increasing engine speed. At low speeds, the longer residence time of the inducted mixture in the intake manifold increases the heating effect and results in lower density mixture and therefore lower volumetric efficiency. In addition to the

heating effect, the late closure of intake valve, which is beneficial at high speeds, causes a backflow of the inducted mixture at lower speeds decreasing the volumetric efficiency. At high speeds, the volumetric efficiency drops due mainly to the sharp increase in flow friction. The decrease in volumetric efficiency has an adverse effect on the engine output since less mixture will be available for combustion.

The percent of input energy lost in heat transfer to the cylinder walls is substantial at lower speeds due to the longer time of contact between the combustion gases and the cylinder walls. The greater heat losses at lower speeds result in lower thermal efficiency and therefore higher specific fuel consumption. The fraction of engine power required to overcome the mechanical friction between the engine moving parts increases sharply with engine speed and becomes substantial at high speeds decreasing the brake thermal efficiency.

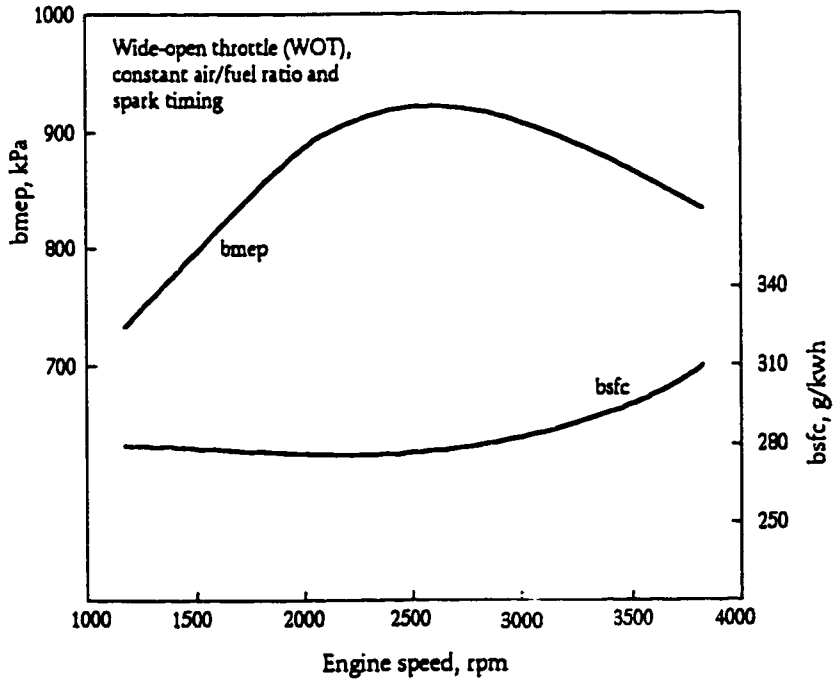

Figure 2-5: Variation of brake mean effective pressure (bmep) and brake specific fuel consumption (bsfc) with engine speed at wide-open throttle conditions for a 3.8-liter six-cylinder spark-ignition engine [2].

The described variations in engine performance characteristics with speed affect the formation mechanisms of NOx, HC, and CO emissions. As the engine speed increases, the fraction of the residual gas left over from the previous cycle decreases and the fraction of energy lost in heat transfer also decreases resulting in higher temperatures during combustion, expansion, and exhaust processes. The higher combustion temperature increases the NOx concentration while the higher expansion and exhaust temperature promotes the post-reaction of unburned fuel and thus decreasing the HC emissions. The load has a similar effect as that of speed since the increase in load increases the combustion, expansion, and exhaust temperatures. Also, the increase of engine speed increases the gas turbulence inside the combustion chamber and therefore improves the mixing quality between air and fuel. This improved mixing results in better combustion and therefore less unburned or partially burned fuel. In other words, increasing the engine speed results in less HC emissions. In the case of CO emissions, the positive effect of improved mixing and higher exhaust temperature is offset by the decreasing residence time in the exhaust manifold [2, 11].

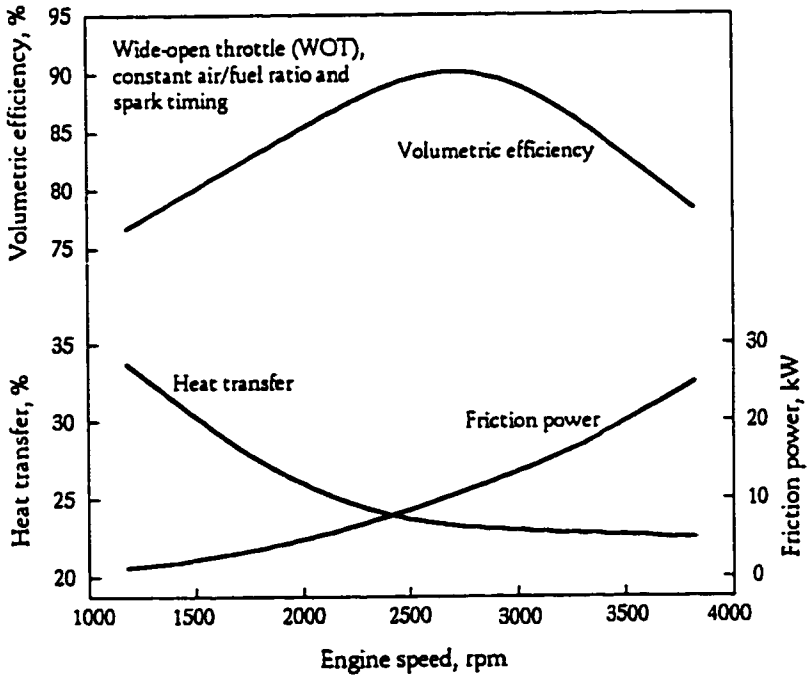

Figure 2-6: Typical variation of volumetric efficiency, fraction of input energy lost in heat transfer, and friction losses with engine speed [2].

2.2 Role of Oxygenated Fuel Properties

The addition of oxygenates influences the physical and chemical properties of the fuel. Properties like boiling temperature, vapor pressure, latent heat of vaporization, energy content, flame temperature, and flame speed are affected. These changes in fuel properties are expected to influence the combustion characteristics, and therefore, the engine performance and exhaust emissions [11, 12, 13, 14, 15, 16]. To understand these influences, we need to examine the effects of oxygenates on the original properties of gasoline.

2.2.1 Effect of Stoichiometric Air/Fuel Ratio

Oxygenates have lower stoichiometric air/fuel ratios than gasoline, therefore, blends of gasoline and oxygenates will have lower air/fuel ratios than that of gasoline. The decrease in air/fuel ratio is proportional to the ratio of the oxygenate in the blend. The use of blends of gasoline and oxygenates, as a replacement of gasoline on unmodified engines, will shift the actual equivalence ratio to the lean side of operation. This shift results in what is called the leaning effect. The leaning effect is a result of the existence of oxygen in the oxygenated blend. The shift in equivalence ratio is proportional to the oxygen weight in the blend.

Modern engines operating with unleaded fuels can be equipped with a three-way catalyst (TWC). This catalyst can significantly reduce the concentrations of CO, HC, and NOx emissions provided that the engine operates with a close-to-stoichiometric mixture. Figure 2-7 shows the influence of equivalence ratio on the conversion efficiency of the three-way catalyst. A conversion efficiency of 80 % or more can be achieved in a very narrow range of equivalence ratios (0.998-1.008). This narrow range is called the TWC 80 % efficiency window. Conventional mixture formation systems can not handle this strict range therefore more sophisticated closed-loop feedback control systems are currently applied. Figure 2-8 illustrates the principle of such closed-loop control systems [2, 4].

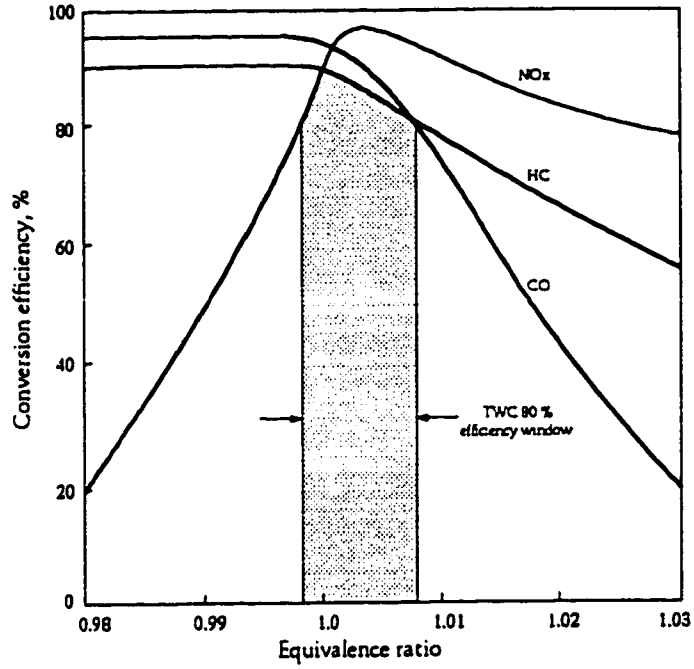

Figure 2-7: The influence of the equivalence ratio on the conversion efficiency of three-way catalysts [2, 4].

The leaning effect will have also an effect on the engine performance. It can be seen from Figure 2-2 that a slight decrease in equivalence ratio from stoichiometric will be advantageous to the thermal efficiency but will decrease the maximum output.

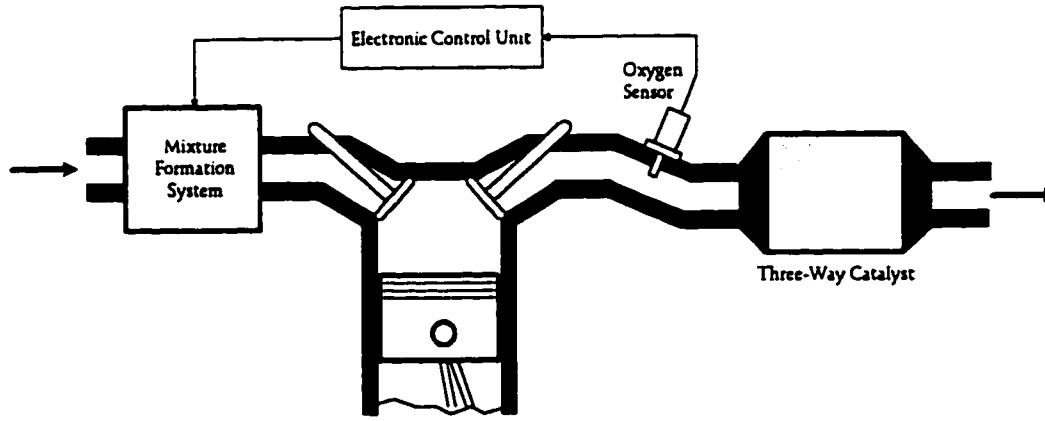

Figure 2-8: A schematic diagram illustrating the principle of closed-loop control of air/fuel ratio used with three-way catalysts [17].

2.1.2 Effect of Heating Value

The heating value of the fuel is the amount of heat released as a result of complete combustion of the fuel. Oxygenates usually have lower heating values than gasoline. The ratios of heating values of MTBE, ethanol, methanol to that of gasoline are 80%, 60%, and 45% respectively. Blends of gasoline and oxygenates have lower heating values than that of gasoline. The decrease in heating value is proportional to the ratio of the oxygenate in the blend. Lower heating value results in less heat release during combustion, and therefore, lower flame temperature. The decrease in flame temperature is expected to influence the formation of nitrogen oxides because of their strong dependence on temperature. In addition, lower heating value means that more fuel is needed to produce same amount of power. In other words, the volume-based fuel consumption will increase [8, 13, 18, 19].

2.1.3 Effect of Latent Heat of Vaporization

The latent heat of vaporization is the amount of heat required to evaporate certain mass of a substance. In the case of SI engines operating with liquid fuels, the fuel is normally introduced, in liquid phase, to the air in the intake manifold before entering the combustion chamber. In order to evaporate, the liquid fuel absorbs the necessary heat from the manifold walls and from the intake air lowering its temperature and thus increasing its density. The increase in air density allows more air to fill the engine cylinders increasing the volumetric efficiency and thus increasing engine output. This role of latent heat of vaporization is usually referred to as the cooling effect. The MTBE has a value of latent heat of vaporization comparable to that of typical gasoline. Methanol and ethanol, on the other hand, have much higher values than that of gasoline. This makes the methanol and ethanol more effective in cooling the intake air and thus producing higher engine output. This is more evident in the case of gasoline fuel injection systems where the atomized liquid fuel is injected to the air just before the combustion chamber allowing most of the heat to be absorbed from the intake air rather

than the intake walls. The cooling effect, however, has a disadvantage during the cold-start conditions. Methanol and ethanol blends will be more difficult to evaporate during cold conditions resulting in lower combustion quality and possibly starting problems particularly with high blending ratios. The lower combustion quality increases the unburned HC emissions.

2.1.4 Effect of Boiling Temperatures

Oxygenates are single compounds that have certain boiling temperatures. Gasoline, however, is a mixture of compounds that have different boiling temperatures. The boiling process of gasoline takes place in temperatures typically ranging from 25°C to 225°C. The addition of oxygenates to gasoline changes the boiling behavior of gasoline causing a distortion in the boiling curve at the point where the added oxygenate starts to boil.

In general, addition of oxygenates decreases the temperature required to evaporate the first 50% of the fuel. This decrease in the boiling temperature of the front end (first 50%) is expected to have a good impact on the cold-start conditions because of better evaporation of fuel. In addition, the better evaporation of fuel improves the combustion resulting in less unburned hydrocarbons. However, this advantage is offset in the case of ethanol and methanol by their high latent heat of vaporization, which makes the evaporation more difficult. In hot conditions, the decreased boiling temperature may result in vapor lock in the fuel lines [8, 11, 12, 14, 16, 20].

2.1.5 Effect of Vapor Pressure

The vapor pressure is a measure of the volatility of the fuel. Reid vapor pressure (Rvp) is the vapor pressure in a closed chamber maintained at 38°C. The Rvp of MTBE is close to that of gasoline. When blended with gasoline, MTBE has a little effect on the original Rvp. Ethanol and methanol have much lower Rvp than that of gasoline. However, blending gasoline with methanol results in a significant increase of Rvp. Blending

gasoline with ethanol also increases the Rvp but to less extent. The maximum increase in Rvp results from the addition of about 3 vol. % of oxygenates.

The effects of changing vapor pressure are similar to the effects of changing the fuel front end boiling temperature. The increase in the vapor pressure is expected to be an advantage in cold-start conditions because of better volatility of the fuel. This results in better evaporation of fuel that improves the combustion producing less unburned hydrocarbons. This advantage is also offset in the case of ethanol and methanol by their high latent heat of vaporization that makes the evaporation difficult. In hot conditions, the increase in vapor pressure increases the evaporation emissions and results in more exhaust unburned hydrocarbons [8, 12, 13, 14, 16].

2.1.6 Effect of Flame Temperature and Burning Velocity

Because of their lower heating values than gasoline, oxygenates have lower adiabatic flame temperatures. MTBE has an adiabatic flame temperature similar to the paraffin compounds that form about 60 vol. % of typical gasoline. Alcohols have relatively lower adiabatic flame temperatures (see Figure 2.9). Blending gasoline with ethanol or methanol is expected to slightly decrease the flame temperature of the combustion [8].

The burning velocities of ethanol and methanol are higher than that of gasoline. A study conducted on mixtures of ethanol-isooctane and methanol-isooctane [21] indicated that the addition of ethanol increased the burning velocity. Adding methanol, on the other hand, resulted in a decrease in burning velocity up to about 50 vol. % of methanol in the blend. After that ratio, the burning velocity started to increase.

The changes in flame temperatures and burning speeds are expected to influence the formation of nitrogen oxides because of their dependence on time and temperature. The lower the flame temperature the lower the concentration of nitrogen oxides. The effect of burning speed on the formation of nitrogen oxides is more complicated since the burning velocity affects both the temperature and time [2, 8, 12].

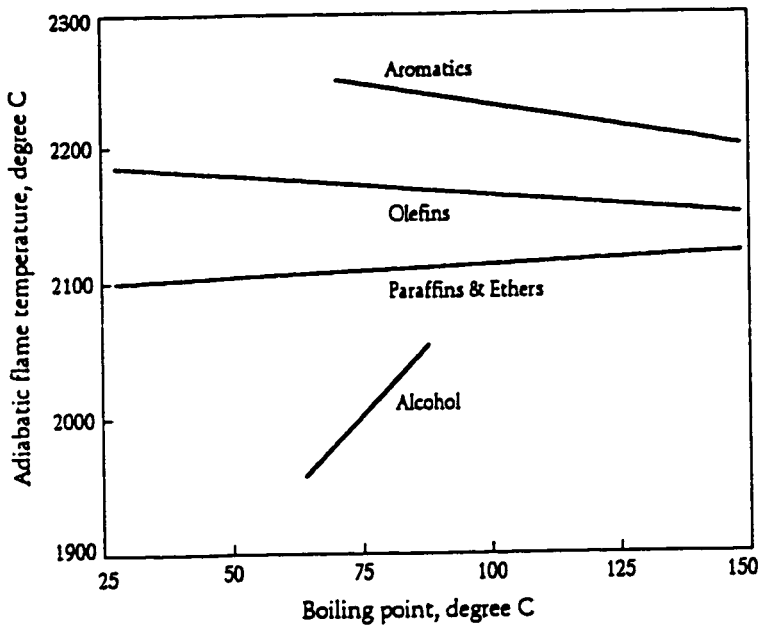

Figure 2-9: The adiabatic flame temperatures for alcohol, ethers, and major gasoline compounds [8].

2.2 Engine Dynamometer Studies

The impact of alternative fuels and additives on output, thermal efficiency, and uncontrolled exhaust emissions of internal combustion engines is usually examined on an engine dynamometer setup. Very well-controlled experiments can be carried out on engine dynamometer setups due to the possible manipulation of almost all the operational variables. Separate effects of oxygenates can be studied by controlling the applicable variables. The engine dynamometer test procedures are very flexible and in many cases are not standardized. US research studies usually conform to the recommended practices of Society of Automotive Engineers (SAE) published in their annual handbook. Exhaust emission measurements normally reflect the engine exhaust concentrations without application of any in-combustion or after-combustion emission control systems. In the following paragraphs, several engine dynamometer studies will be reviewed.

In 1984, Rajan [22] studied the possibility of blending gasoline with ethanol that contains small percentage of water. The tested blends were prepared on the basis of miscibility limits where the composition of ethanol-gasoline mixtures is dependent on the amount of water added. The tests were conducted on an unmodified 2.3-Liter four-cylinder engine equipped with a carburetor. The maximum brake power, thermal efficiency, and NOx and HC emissions were all evaluated at a constant speed of 2050 rpm and two different spark timings (34 and 37.5 degrees btc). These tests were conducted with blends containing up to 7 vol.% water in ethanol. Lower power outputs were realized due to the leaning effect and lower heating value of the blend. The thermal efficiency was found to increase with blends containing up to 4 vol.% water in ethanol, but beyond this limit the efficiency decreased substantially. The NOx emissions continued to decrease appreciably with increasing water percent. This was attributed by the author to the leaning effect. Hydrocarbon emissions were reported to decrease substantially for blends with less than 4 vol.% water in ethanol, but increase sharply beyond that. Advancing the spark timing from 34° to 37° btc was found to increase the output, thermal efficiency, and NOx and HC emissions.

In 1984, Tsao and Li [23] evaluated the performance of a single-cylinder four-stroke engine operating with an oxygenated fuel prepared by blending 30 vol. % methanol with unleaded commercial gasoline. Three mixtures of water-gasoline (5, 10, and 15 vol. % water in gasoline) were prepared, tested, and compared to the base fuel and methanol blend. Using an eddy-current dynamometer, the tests were conducted at a constant speed of 2000 rpm and maximum brake torque timing (MBT). When compared to gasoline, all the blends indicated a gain in thermal efficiency (lower specific energy consumption) with the maximum gain in the case of methanol blend. In addition, the maximum output (at 2000 rpm) increased for the all blends except the 5 vol.% water-in-gasoline. The gain in thermal efficiency and engine output was attributed to the cooling effect of water and methanol on the intake air, and the subsequent improvement in volumetric efficiency. A substantial increase was reported in CO emissions with all the

tested blends, and in HC emissions with the 10 and 15 vol.% water-in-gasoline blends. For methanol and 5 vol.% water-in-gasoline blends, HC emissions decreased.

In 1987, Wang, Yang, and Feng [24] studied the effects of MTBE on performance and emissions of two unmodified Volkswagen 1.8-liter engines. The performance of a test fuel, prepared by adding 14 vol.% MTBE and 6 vol.% alkylate to naphtha, was tested with the two engines on a hydraulic dynamometer. The researchers reported a slight increase in the engine maximum torque particularly at medium and high speeds, and a slight decrease in the specific fuel consumption. CO and HC emissions were reported for a 13 vol.% MTBE blend. A significant reduction in CO and HC emissions was reported as compared to the emissions from a reference fuel. The researchers attributed the decrease in CO and HC emissions to the shift in stoichiometric air/fuel ratio towards the lean side (i.e. the leaning effect).

In 1989, Bata and Roan [25] studied the effects of alcohol-gasoline blends on exhaust emissions from a 2.3-Liter four-cylinder engine. Fuel blends with different ratios (5, 10, 15 and 20 vol.%) of methanol and/or ethanol in gasoline were tested. The tests were carried out at a constant speed (2500 rpm) and a constant load. Exhaust emissions were measured at different equivalence ratios (0.85, 1.0, and 1.15) and different spark timings (10, 20, and 30° btc). A significant reduction in CO emissions, particularly on the lean side near stoichiometric and with higher ratios of alcohol in the blend, was reported for all the blends. The spark timing had insignificant effect on the production of CO emissions. The reduction of CO emissions with alcohol blends was attributed partly to the cooling effect of alcohol (due to its higher latent heat of vaporization). The HC emissions slightly increased with the minimum increase observed at an equivalence ratio on the lean side. According to the authors, the increase in HC emissions was due to the higher vapor pressure of alcohol blends. The researchers found that, under the same conditions, the type of alcohol was insignificant on the production of CO and HC emissions.

In 1991, Rice et al. [26] studied the exhaust emissions of alcohol-gasoline blends. They conducted their tests on a 2.2-Liter four-cylinder engine coupled to a hydraulic dynamometer and operated with blends comprised of 20 vol.% of ethanol, methanol or butanol in gasoline. The effect of equivalence ratio (0.8-1.2) on formation of CO, unburned HC, and NOx emissions was examined at an engine speed of 1500 rpm and a load of 125 Nm. A reduction in CO emissions was observed for all alcohol blends relative to gasoline, particularly for fuel-rich operation ($\Phi > 1.0$). On a volume basis, unburned fuel emissions were higher for alcohol blends, but on mass basis, only small differences were noted. At very lean conditions, the methanol blend had a more pronounced increase in unburned HC emissions that might reflect, according to the authors, a misfire tendency. For fuel-rich operation ($\Phi > 1.0$), the NOx emissions from all alcohol blends were appreciably lower. However, for fuel-lean operation ($\Phi < 1.0$), NOx emissions were comparable to that of gasoline in the case of ethanol and butanol blends and greater in the case of methanol blend. The reduction in NOx emissions achieved with alcohol blends was attributed to the lower heating values and flame temperature.

In 1994, Kowalewicz [27] investigated the influence of methanol content in methanol-gasoline blends on engine thermal efficiency. The experiments were conducted on a 1.5-Liter engine with the fuel system modified so that the air/fuel ratio could be controlled. The tested blends were 10, 20, 40, and 60 vol.% methanol in gasoline. The engine efficiency was examined at different speeds (2000-3500 rpm) and different loads (5-105 Nm). The air/fuel ratio was kept constant for each speed but equivalence ratio varied with fuels. No adjustment was made to the spark timing during the whole tests. The researcher reported an increase in engine efficiency as the methanol content in the blend increases. The increase in thermal efficiency was attributed to the lower combustion temperatures which result in lower compression pressures and therefore lower compression work. The lower combustion temperatures result also in lower heat losses. The increase in thermal efficiency was also attributed to the increase in expansion work resulting from the increase in volume of moles of alcohol combustion products.

Emissions of CO and HC were reported to be lower for a higher methanol content in the blend.

2.3 Chassis Dynamometer Studies

In order to evaluate the overall impact of using oxygenated fuels on new or currently running vehicles, chassis dynamometer tests are usually employed. Chassis dynamometer tests involve testing the performance of the vehicle as a whole rather than the engine. The drive wheels of the vehicle are placed on the rollers of the chassis dynamometer that simulate the road driving conditions. In order to obtain reliable and reproducible results, a precisely defined program is usually carried out. This program is usually described by the legislation authorities in each country as a tool to test the conformity of vehicles to local emission regulations. A typical program would have driving cycles that represent actual driving conditions including cold-starting, warming-up, acceleration, and constant speed conditions, and would usually involve examining regulated exhaust emissions and fuel economy. Exhaust emissions are usually measured using the Constant Volume Sampling (CVS) method in which the exhaust emissions are continuously diluted with a proportional amount of air. During the test program, a certain percentage of the diluted exhaust emissions is continuously collected in bags. The contents of these bags are analyzed upon the conclusion of the test so that the analysis results represent the average for the whole test. Fuel economy is calculated in terms of driving distance over the volume of the consumed fuel. US research studies conducted on chassis dynamometer usually follow the US EPA's (Environmental Protection Agency) 1975 Federal Test Procedure which is composed of three phases of speed-time curves based on actual morning rush-hour driving cycles. In the next paragraphs, several chassis dynamometer studies will be discussed. It is important to emphasize that these studies are conducted on unmodified vehicles, and the exhaust emissions reported were the tailpipe emissions which might have been treated with catalysts or other different emission control technologies. No adjustment or control was applied on any engine

parameter during the test. For these reasons, the results from these studies give an overall statistical indication of the response of unmodified vehicles to the application of the tested fuels.

In 1974, Wigg and Lunt [34] evaluated the effects of methanol on fuel economy and exhaust emissions of three carburetor-equipped vehicles. Tests were carried out according to the Federal Test Procedure (FTP). The test fuels were a typical unleaded gasoline (as the base), a 15 vol.% methanol blended with the base, and a 15 vol.% methanol blend with adjusted volatility to match that of the base. In the case of adjusted-volatility blend used on older rich-operating ($\Phi>1.0$) vehicle, the researchers reported a significant improvement in energy-based fuel economy and a decrease in CO and HC emissions, but an increase in NOx emissions. The results for non-adjusted volatility blends indicated less improvement in fuel economy and exhaust emissions. Newer mostly lean-operating ($\Phi<1.0$) vehicles showed no significant benefits with methanol blends. Both the fuel economy and exhaust emissions results were explained in the context of the leaning effect. The study also pointed to some drivability problems of methanol/gasoline blends, such as vapor lock and phase separation. A noticeable increase in vapor lock tendency was observed when no front-end volatility adjustments were made to the methanol blends. The phase separation was also found to be a potential problem.

In 1975, Brinkman, Gallopoulos and Jackson [35] studied the effect of blending methanol and ethanol with gasoline on the exhaust emissions, fuel economy, and drivability. They tested a total of 16 vehicles according to the US Federal Test Procedure. They found that adding 10 vol.% methanol to unleaded gasoline without altering the fuel system decreased CO emissions, volume-based fuel economy, and drivability. On the other hand, the addition of methanol had conflicting trends (increase, decrease, or no effect) on the total HC and NOx emissions depending on the composition of base fuel used and the vehicle tested. They also reported that ethanol and methanol behaved

similarly. The researchers attributed these results to the lower stoichiometric air/fuel ratio, lower heating value, higher volatility, and lower combustion temperature of alcohol/gasoline blends.

In 1980, Furey and King [28] measured exhaust emissions from vehicles fueled with gasoline containing 15 vol.% MTBE or 10 vol.% ethanol. They conducted the tests on three vehicles using the US Federal Test Procedure. The base fuel, to which the results were compared, was unleaded indolene. For both blends, exhaust HC, CO, and NOx emissions from vehicles without closed-loop fuel systems were reduced significantly. For vehicles equipped with closed-loop carburetors, adding the oxygenates made no significant effect on exhaust emissions. They reported also a decrease in the volume-based fuel economy and a deterioration in cold-start and warmed-up drivability with oxygenated blends. The researchers attributed the decrease in exhaust emissions and the deterioration in drivability to the leaning effect, and the decrease in volume-based fuel economy to the lower heating values of the blends.

In the same year, Naman and Allsup [29] studied the effect of MTBE, methanol, and ethanol on the exhaust emissions and fuel economy. They tested eight 1978 vehicles according to the US Federal Test Procedure. Five of these vehicles were equipped with oxidation catalyst and the rest with three-way catalysts. For the vehicles equipped with three-way catalysts and closed-loop emission control systems, the researchers found that the addition of all the oxygenates to the base fuel (indolene) decreased CO and HC emissions but increased NOx emissions. For vehicles with oxidation catalyst, the CO and HC emissions decreased with all the blends and NOx emissions increased in case of methanol and remained almost unchanged in the case of MTBE and ethanol blends. For all the vehicles, the volume-based fuel economy decreased.

In 1989, Gething, Welstand and Horn [30] studied the relationship between the reduction in vehicle CO exhaust emissions and the fuel oxygen content. They carried out their tests according to US Federal Test Procedure on 18 vehicles (1968 to 1988

models) with different emission control systems. Two blends were prepared by adding 11 vol.% MTBE (2 wt.% oxygen) and 10 vol.% ethanol (3.5 wt.% Oxygen) to regular unleaded gasoline which was used as a base fuel. The researchers found that the reduction in carbon monoxide due to using oxygenated fuel varies from vehicle to another. For vehicles without catalyst, the CO reduction was proportional to fuel oxygen content while it was not proportional in the case of vehicles equipped with oxidation or three-way catalyst systems. Furthermore, for vehicles with three-way catalyst and adaptive learning systems (systems that correct their mixture formation strategies according to the most recent operating conditions), the researchers reported a very little response to the oxygenated fuels. They reported also that, in the case of vehicles equipped with oxidation and three-way catalysts, the MTBE blend with 2.0 wt.% oxygen was 80% as effective in reducing the CO emissions as the ethanol blend with 3.5 wt.% oxygen.

In 1991, Hochhauser et al. [31], as a part of the cooperative research program (AQIRP) initiated by several U.S. automobile and oil companies, measured the mass exhaust emissions form current vehicles (twenty 1989 model) and older vehicles (fourteen 1983-1985 model). Eighteen fuels with variable composition and MTBE content were tested according to the Federal Test Procedure. One of the parameters they measured was the effect of MTBE content (0-15 vol.%) in gasoline. They found that adding MTBE reduced HC and CO emissions. In the case of NOx emissions, the researchers observed a significant interaction between the effects of MTBE and aromatics content of fuels. Adding MTBE raised NOx for lower aromatics fuels, but had no significant impact for higher aromatics fuels.

In 1992, Reuter et al. [32], also as a part of the cooperative research program (AQIRP), studied the effects of oxygenated fuels and Reid vapor pressure (Rvp) on exhaust emissions. Eleven fuels with different vapor pressures and oxygenate contents were used. The tested oxygenates were MTBE (15 vol.%), ETBE (17 vol.%) and ethanol (10 vol.%).

The tests were conducted according to the US Federal Test procedure. Exhaust emission results indicated that a reduction in fuel Reid vapor pressure reduced exhaust HC and CO but increased NOx emissions. The addition of oxygenates to gasoline was found to decrease exhaust HC and CO but generally increased NOx emissions. No significant differences between oxygenates were identified.

In 1994, Stump et al. [33] measured exhaust emissions from three pre-1985 passenger motor vehicles operating on an MTBE-oxygenated (9.5 vol.%) and on a non-oxygenated regular unleaded fuel. Emissions data were collected according to the Federal Test Procedure at different room temperatures to simulate ambient driving conditions. The study showed conflicting effects of MTBE on CO, HC, and NOx emissions for different vehicles and different emission control systems. But in general, as the test temperature increased, exhaust HC and CO emissions decreased and NOx emissions increased for both fuels and all vehicles.

In 1996, Stump et al. [36] measured exhaust emissions from three pre-1985 passenger motor vehicles operating with a regular unleaded fuel oxygenated with 8.8 vol.% ethanol. The tests were carried out according to the US Federal Test Procedure, and the results were compared to those of the base fuel. The vapor pressure of the prepared blend was adjusted to match that of the base fuel. Emission data were collected at 90 °F, 75 °F, and 40 °F to simulate ambient driving conditions. The study showed that ethanol blend generally reduced exhaust total HC and CO, but increased NOx emissions. However, the trends showed variation with different ambient temperatures and with different vehicles that contain different emission control technologies. Exhaust total HC and CO emissions from both fuels and all vehicles decreased as the test temperature increased. On the other hand, NOx emissions increased as test temperature increased. The increase in HC and CO emissions at lower operating temperatures was attributed to the fuel enrichment during cold conditions.

Table 2-1: Summary of the experimental research on the effects of MTBE, methanol, and ethanol on exhaust emissions.

Reference	Blend	Notes	CO	HC	NOx
Rajan [22]	E-water-gasoline	< 4% water in E	-----	⇩	⇩(33-47%)
Tsao and Li [23]	M-30 (30 vol. %)	Single cyl. - 2000 rpm	⇧(42%)	⇩(3%)	-----
Bata and Roan [25]	E, M-(0-20)	Carbureted engine	⇩(40-50%)	⇳ (variable spark time)	-----
Rice et al. [26]	E-20, M-20 (ϕ=1)	Four-cyl. engine	⓪	-----	⇩(9-16%)
Kowalewicz [27]	M-(0-60)	Two Carbureted engines	⇳ (variable load)	-----	-----
Furey and King [28]	MTBE-15	CL	⓪	⓪	⓪
		OL	⇩(28%)	⇩(18%)	⇩(20%)
	E-10	CL	⓪	⓪	⓪
		OL	⇩(26%)	⇩(18%)	⇩(20%)
Naman & Alsup [29]	MTBE-7	OC	⇩(32%)	⇩(20%)	⓪
		TWC-CL	⇩(28%)	⇩(28%)	⇧(35%)
	M-10	OC	⇩(46%)	⇩(27%)	⇧(16%)
		TWC-CL	⇩(28%)	⇩(5%)	⇧(48%)
	E-10	OC	⇩(46%)	⇩(32%)	⓪
		TWC-CL	⇩(27%)	⇩(9%)	⇧(23%)
Gething et al. [30]	MTBE-11	NC, OC-OL, TWC-CL	⇩(18-25%)	-----	-----
		TWC-CLAL	⓪		
Hochhauser et al. [31]	MTBE-(0-15)	Twenty 1989 cars	⇩ (11%)	⇩ (5.5%)	⓪
		Fourteen 1983-85 cars	⇩ (14%)	⇩ (9%)	⓪
Reuter et al. [32]	MTBE-15	Twenty 1989 cars	⇩(9.3%)	⇩(6.5%)	⇧(3%)
	M-10		⇩(13.4%)	⇩(4.9%)	⇧(4.5%)
Stump et al. [33]	MTBE-9.5	Three pre-1985 cars	⇩(5-26%, 40°F intake)	⇩(6-40%, 40°F intake)	⇳ (40°F intake temp)
Brinkman et al. [35]	M-10	Two cars (carburetor)	⇩(20-43%)	⇳	⇳
Stump et al. [36]	E-8.8	Three pre-1985 cars	⇩(20-40%, 40°F intake)	⇩(12-48%, 40°F intake)	⇧ (0-50%, 40°F intake)

E : ethanol, **M** : methanol, ⓪ : no or slight effect, ⇧ : increase, ⇩ : decrease, ⇳ : conflicting trends, **OL** : open-loop control, **NC** : no catalyst
OC : oxidation catalyst, **RC** : reduction catalyst, **TWC** : three-way catalyst, **CLAL** : closed-loop with adaptive learning control, **CL** : closed-loop control

Chapter
3

Experimental Set-up

3.1 Background

The experiments were designed to evaluate the effects of oxygenates on performance and exhaust emissions of a spark-ignited internal combustion engine. The performance was evaluated in terms of the maximum engine output and the specific fuel consumption. The exhaust emissions measured were the carbon monoxide (CO), total hydrocarbons (THC), and the nitrogen oxides (NOx).

Measuring the engine output normally involves applying an external load on the crankshaft of the engine. This is normally accomplished using a device called a dynamometer. The dynamometer acts like a braking device that absorbs the engine energy. Frictional, hydraulic, or magnetic forces can achieve the dynamometer braking effect. The fuel consumption measurement involves determination of the rate of fuel flow to the engine. Several approaches are used to determine the fuel flow rate, but the simplest is to use calibrated volumes with a stopwatch. More complicated approach is to utilize a gravimetric metering unit in which the weight of the fuel, supplied to the engine over a certain interval, is directly measured.

The exhaust emission concentrations are measured using specialized analyzers. The carbon monoxide (CO) is measured by the *nondispersive infrared* (NDIR) analyzer. This analyzer operates on the basis that the CO absorption of a certain wavelength of infrared radiation is proportional to the concentration of CO in the exhaust sample. The concentration of total hydrocarbons (THC) is measured by the *flame ionization detector* (FID). The FID analyzer is built on the fact that passing a gas sample containing hydrocarbons through a hydrogen flame causes a current of ions, proportional to the amount of carbon atoms present in the sample, to flow. The FID analyzer needs to be

properly configured to work with samples that contain substantial amounts of oxygen, such as the exhaust samples from diesel or alcohol fueled engines. This is because the presence of oxygen in the sample affects the FID response by inhibiting the production of ions. Applying certain modifications to the analyzer can minimize the effect of oxygen. The concentration of nitrogen oxides (NOx) is measured by the *chemiluminescent* (CL) analyzer which is based on the fact that the reaction between nitric oxide (NO) and ozone (O_3) emits photons of light proportional to the concentration of (NO). The nitrogen oxides consists mainly of nitric oxide (NO) and a small amount (maximum 2 % for SI engines) of nitrogen dioxide (NO_2). The nitrogen dioxide present in the sample is converted to nitric oxide prior to the reaction with ozone.

3.2 Test Facilities

Figure 3-1 shows a schematic of the testing facility used in this study. The six-cylinder test engine is manufactured by Mercedes-Benz and has a swept volume of 2960 cm^3. It has a bore of 88.5 mm, a stroke of 80.2 mm, a compression ratio of 9.2, and a maximum power of 132 kW at 5700 rpm. The engine is equipped with the KE-Jetronic continuous fuel injection system. This is a multi-point injection system where the fuel is continuously injected through injection valves directly before the intake valve of each cylinder. The engine has an electronic ignition system with an electronic spark timing adjustment. The temperatures of cooling water and lubrication oil are controlled by two fitted heat exchangers.

The engine is coupled to an eddy-current dynamometer. The braking effect of this dynamometer is achieved through a magnetic field created by a supplied electrical current. To rotate through the magnetic field, the revolving part of the dynamometer that is connected to the engine dissipates part of the engine energy in form of heat. The dissipated engine energy is proportional to the strength of the magnetic field that is controlled through the supplied current. This eddy-current dynamometer is

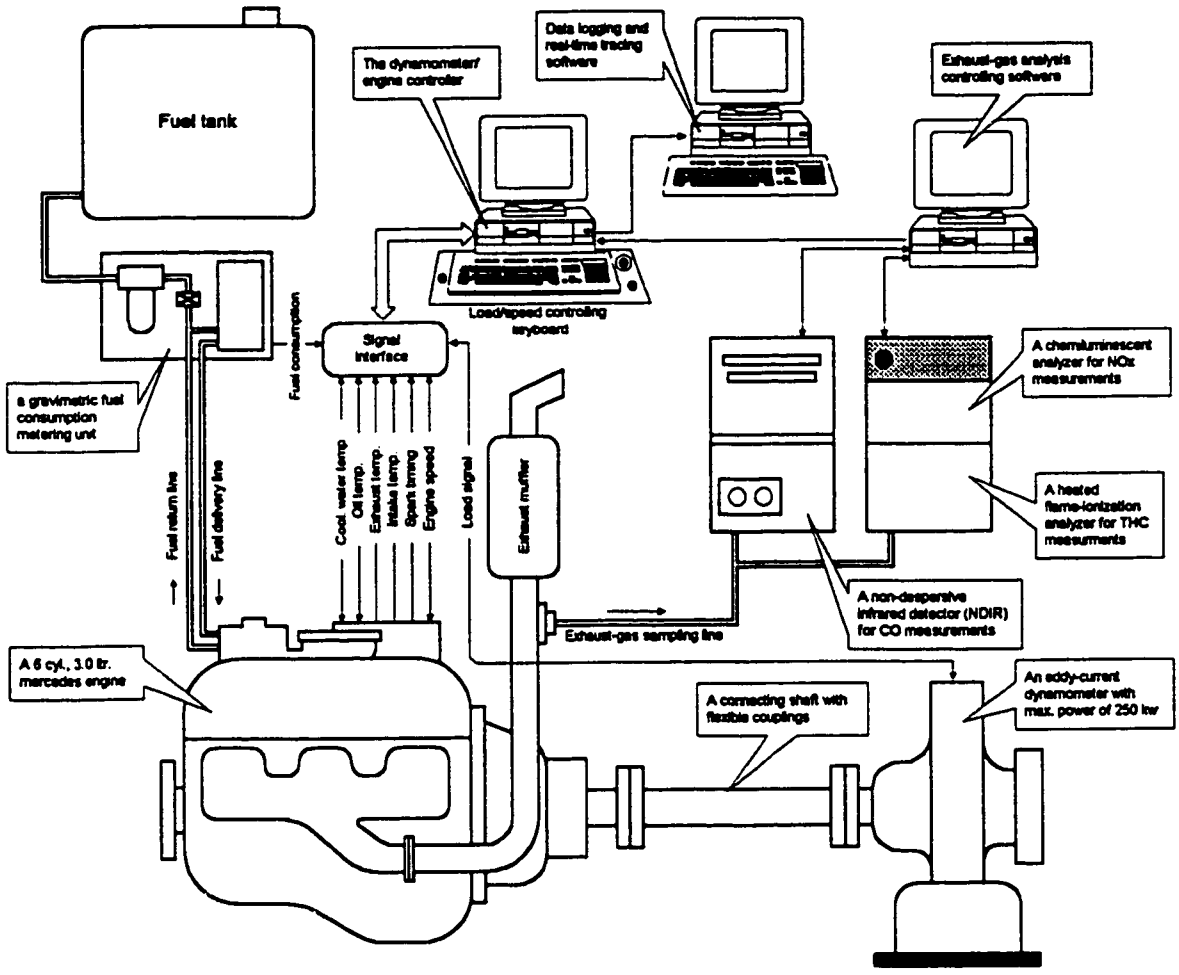

Figure 3.1: A schematic diagram illustrating the major testing facilities.

electronically controlled and water-cooled. It has a maximum power of 257 kW, a maximum torque of 1400 Nm, and a maximum speed of 8000 rpm.

The dynamometer and the test engine are controlled by a microcomputer equipped with a high-speed data acquisition and logging system. This controlling system receives operational data from various sensors fitted on the engine and the dynamometer. These sensors measure different parameters such as engine load, engine speed, spark timing, intake air temperature, exhaust gas temperature, oil temperature, and cooling water temperature.

Several actuators are also fitted on the engine to execute the controller commands. Examples of these actuators are the throttle actuator and the pneumatic valves that control the oil and cooling water temperatures. The controller also receives the fuel consumption data from a gravimetric fuel consumption metering unit. All the data received are displayed on the controller monitor and at the same time logged to another microcomputer to be stored and traced over time.

The concentration of carbon monoxide (CO) is measured using a Horaiba AIA-722 nondispersive infrared (NDIR) analyzer. The measurement range of this analyzer is 5000 ppm-10 vol. %. A Horaiba FIA-725A heated flame ionization detector (HFID) measures the concentration of the total hydrocarbons (THC). This analyzer is configured to work with exhaust samples of gasoline, diesel, or alcohol fueled engines. The measurement range is 10 - 50000 ppm C. The concentration of nitrogen oxides (NOx) is measured by a Horaiba CLA-720M chemiluminescent (CL) analyzer which has a measuring range of 10-10000 ppm. For the three analyzers, the span repeatability is less than ±1% of the relative scale, and the span drift is less than ±1% of the full scale over a 24-hour period. The measurement results of the three analyzers are displayed on the screen of the exhaust-gas analysis controller and at the same time transferred to the engine/dynamometer controller through a local area network (LAN) cable.

3.3 Test Fuels

The aim of the study is to investigate the effects of oxygenates on the performance and emissions of a gasoline engine. A *base* fuel was prepared by mixing 20 % of naphtha with 80 % of reformate on volumetric basis. A *leaded* fuel was prepared by adding tetra ethyl lead (TEL) to the base fuel. The addition of TEL brought the lead concentration in the fuel to the current maximum allowable level of 0.4 g pb/Liter. The tested oxygenates are the MTBE (methyl tertiary butyl ether), methanol, and ethanol. The chemical structures of these three oxygenates are shown below in Figure 3-2. Each one of the three oxygenates is blended with the base fuel in three ratios: 10, 15, and 20 vol. %. The

MTBE/base blends were designated MTBE10 (10 vol. % MTBE + 90 vol. % base), MTBE15, and MTBE20. The methanol/base blends were designated METH10, METH15, and METH20. The ethanol/base blends were designated ETH10, ETH15, and ETH20.

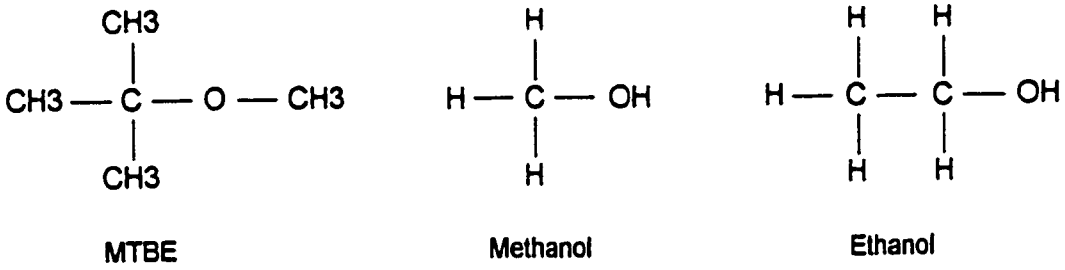

Figure 3-2: The chemical structures of MTBE, methanol, and ethanol

The naphtha and the reformate were obtained from Ras Tanura refinery of Saudi Aramco. The MTBE and methanol were obtained from the National Methanol Company (Ibn Sina) in Jubail, Saudi Arabia. The purity of the MTBE was 98.71 wt. %, and the purity of methanol was 99.99 wt. %. The ethanol was obtained from Saudi Petrochemical Company (SADAF). This was a crude ethanol of 91.0 wt. % purity and contains 7.8 wt. % of water. The presence of water is known to affect the miscibility of alcohol in gasoline. A blend of alcohol and gasoline tolerates the presence of water up to certain percentage, then any additional water will cause the alcohol to separate from the gasoline. The start of phase separation is characterized by formation of haze or cloudiness in the blend, and defined as the cloud point. The cloud point depends on the alcohol content of the blend, the aromatics content of the gasoline, and the temperature. Ethanol is known to have appreciably more tolerance to water than methanol. Figure 3-3 shows the influence of ethanol content, the aromatics content, and the temperature on the water tolerance of ethanol/gasoline blends.

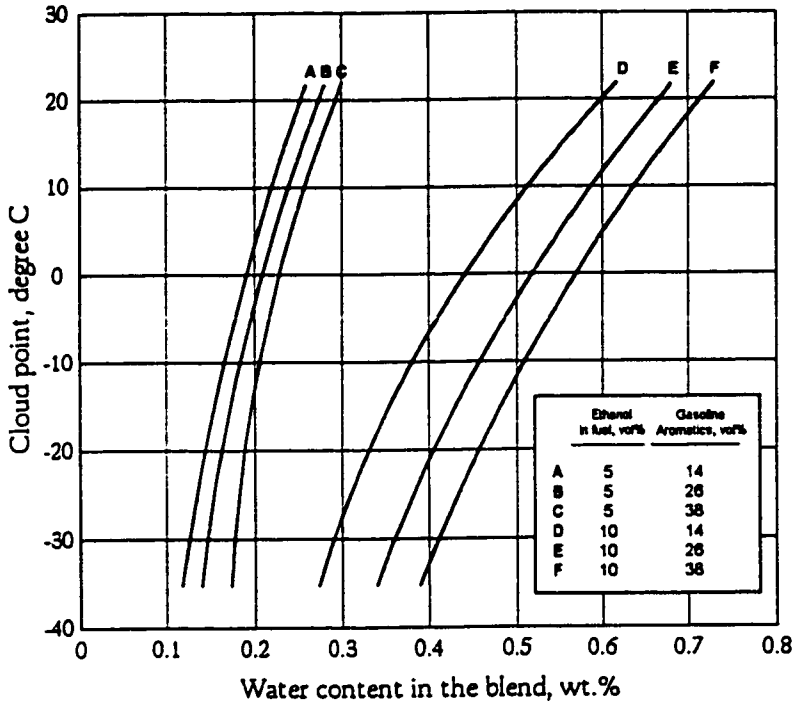

Figure 3-3: Influence of ethanol content, aromatics content, and temperature on the water tolerance of ethanol/gasoline blends [37].

In the case of the ethanol/base blends prepared in the current study, the haze formation, or cloud point, was encountered only with the ETH10 blend (10 vol. % ethanol + 90 vol. % base) at a room temperature of about 20 °C. No signs of phase separation were noticed beyond that temperature. In order to avoid the phase separation when operating with the ethanol/ base blends, a circulation pump was fitted in the fuel tank to ensure continuous mixing of the fuel.

The relevant physical and chemical properties of the test fuels were determined. Tables 3-1 and 3-2 list those properties for the oxygenates and the test fuels respectively. Some of the listed properties were not measured but rather obtained from the cited literature. The distillation characteristics for the blends are shown in Figures 3-4, 3-5, and 3-6. The measurements of specific gravity, Rvp, and distillation characteristics were conducted at the Oil Testing Center of the KFUPM department of chemistry. The rating

of octane number was carried out in the Heat Engine Laboratory of the KFUPM mechanical engineering department. The analysis of test fuels to determine the hydrocarbon types was conducted at the KFUPM research institute.

Table 3-1: Properties of the oxygenates tested in this study.

Property	MTBE	Methanol	Crude ethanol
Chemical formula	$C_4H_9OCH_3$	CH_3OH	$C_2H_5OH + 0.2162\ H_2O$
Weight percent oxygen	18	50	39
Reid vapor pressure (kPa)	61.2	37.1	17.5
Boiling temperature, °C	53.3	63.5	75.6
Specific gravity @ 15.56 °C	0.7461	0.7954	0.816
Research octane number	116 *	133 *	129 *
Lower heating value (MJ/kg)	35.2 **	20.0 **	24.77 ***
Latent heat of vaporization (kJ/kg)	340 **	1160 **	950 ***
Stoichiometric A/F ratio	11.76	6.47	8.3 ****

* Obtained from SAE handbook [9] for pure MTBE, methanol, and ethanol
** Obtained from Ref. [2]
*** Obtained from Ref. [2] for ethanol and adjusted for the presence of water
**** Calculated for ethanol and adjusted for the presence of water.

3.4 Test Conditions and Procedure

The engine maximum output and the specific fuel consumption were measured for all the prepared blends. The engine exhaust emissions were also measured at a wide range of engine operational conditions. These measurements can be affected by many engine operational parameters such as the spark timing, air/fuel ratio, and cooling water temperature. It can be also affected by the test room conditions such as temperature, atmospheric pressure, and humidity.

3.4.1 Spark Timing

As has been discussed earlier, the spark timing refers to the crank angle where the spark is initiated to start the combustion process inside the combustion chamber. There exists a certain spark timing, called the maximum brake torque timing (MBT), at which the

Table 3-2: Properties of the test fuels.

Fuel Property	Reformate	Naphtha	Base fuel	Leaded	MTBE10	MTBE15	MTBE20	METH10	METH15	METH20	ETH10	ETH15	ETH20
Specific gravity @ 15.56°C	0.7776	0.7293	0.7697	0.7717	0.7638	0.7633	0.7628	0.7692	0.7697	0.7707	0.7747	0.7752	0.7771
Rvp, kPa	43.3	39.2	35.0	33.6	41.0	41.6	42.4	57.4	61.6	66.5	59.5	58.1	54.6
Research octane number	92.7	43	84.7	92	87.9	89.8	91.7	88.2	91.6	94.4	88.4	91.5	93.4
Heat of vaporization, kJ/kg *	----	----	350	350	349	349	348	428	466	505	413	445	476
Weight percent oxygen *	0	0	0	0	1.77	2.66	3.55	5.15	7.71	10.27	4.08	6.10	8.11
Heating value, MJ/kg *	----	----	44	44	43.144	42.715	42.283	41.528	40.298	39.073	41.974	40.969	39.971
Stoichiometric A/F *	----	----	14.6	14.6	14.33	14.19	14.05	13.77	13.35	12.94	13.95	13.62	13.3
Hydrocarbon types, vol. % **													
N-Paraffins	6.64	17.36	8.78	8.78	7.91	7.47	7.03	7.91	7.47	7.03	7.91	7.47	7.03
Iso-Paraffins	39.5	59.98	43.60	43.60	39.24	37.06	34.88	39.24	37.06	34.88	39.24	37.06	34.88
Naphthenes	1.55	12.97	3.83	3.83	3.45	3.26	3.07	3.45	3.26	3.07	3.45	3.26	3.07
Aromatics	50.91	9.05	42.54	42.54	38.28	36.16	34.03	38.28	36.16	34.03	38.28	36.16	34.03

* Typical or calculated values and, if applicable, adjusted for the presence of added oxygenate

** Analysis was conducted on reformate and naphtha then calculated and, if applicable, adjusted for the presence of oxygenates

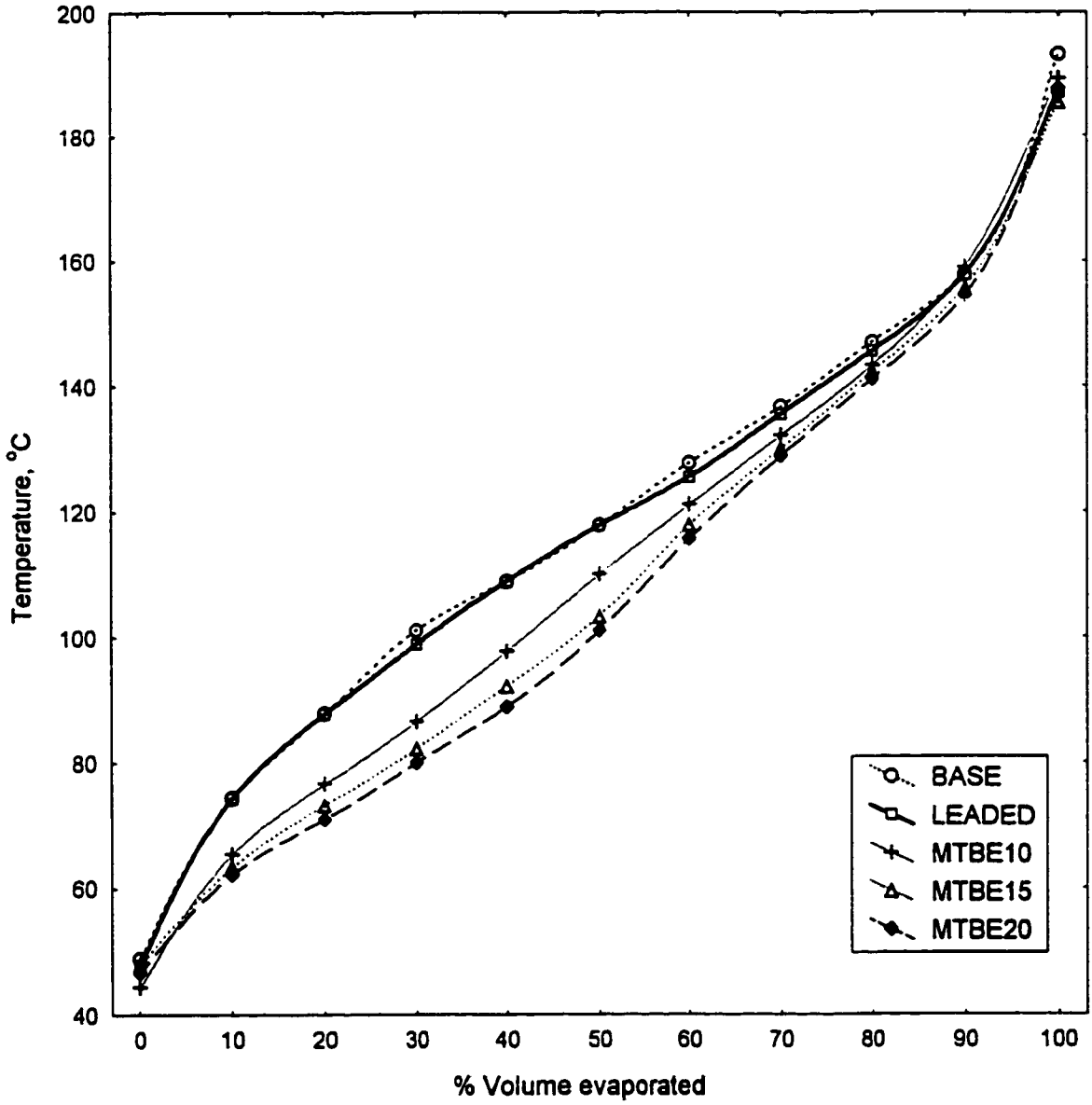

Figure 3-4: D-86 distillation curves for the MTBE blends compared to the base and leaded fuels.

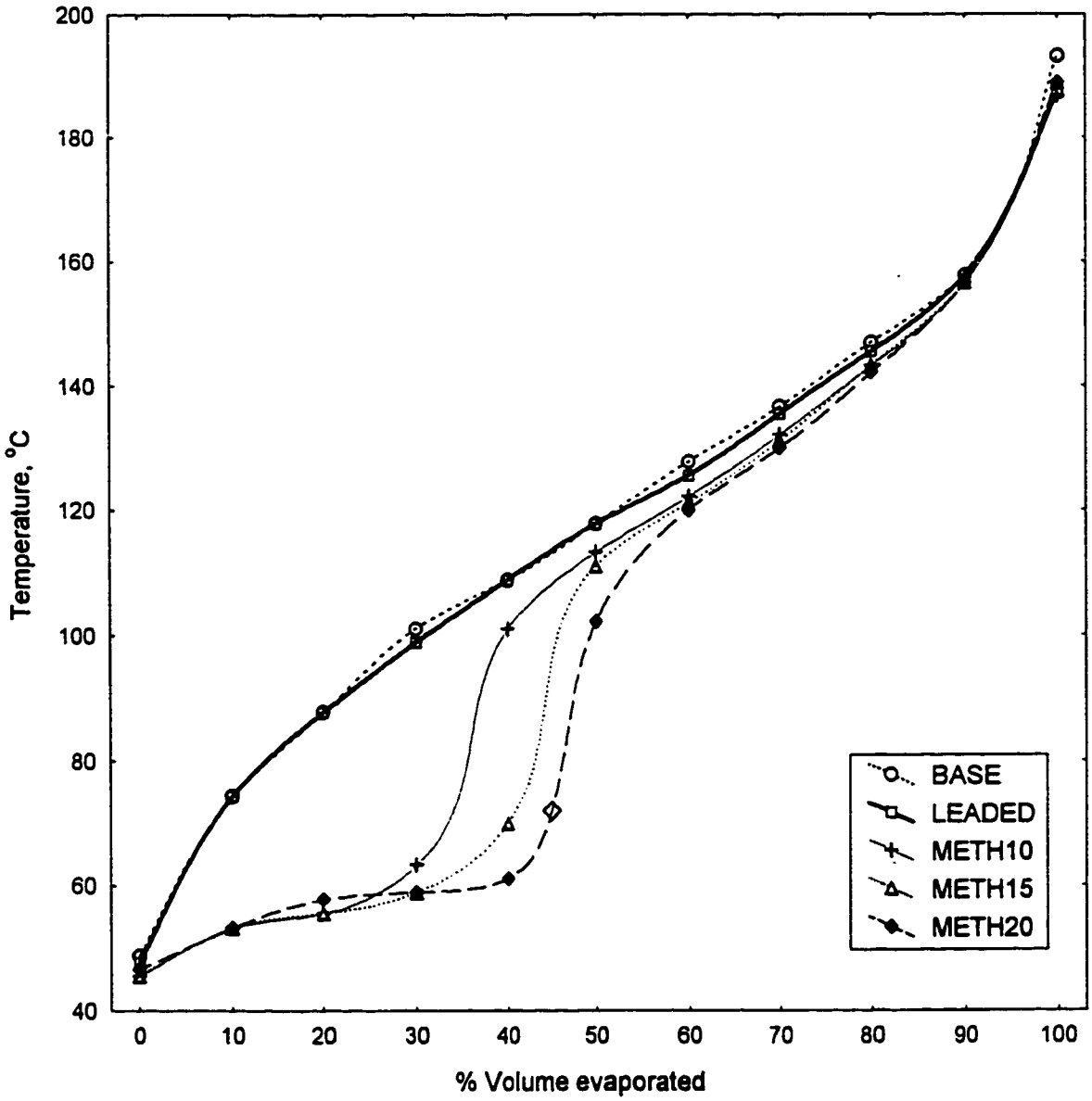

Figure 3-5: D-86 distillation curves for the methanol blends compared to the base and leaded fuels.

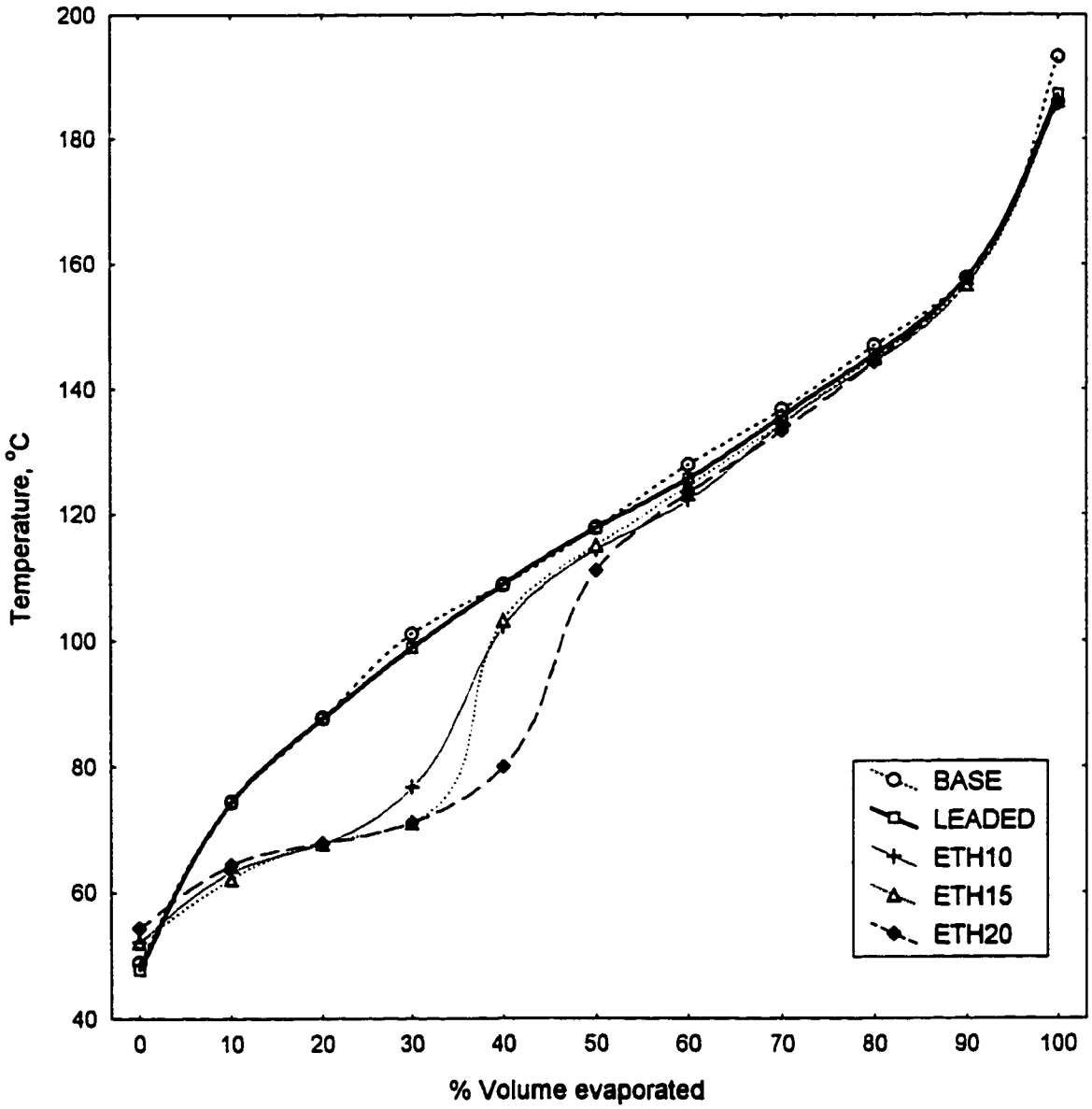

Figure 3-6: D-86 distillation curves for the ethanol blends compared to the base and leaded fuels.

engine performance is optimum. This spark timing gives maximum brake torque and maximum brake thermal efficiency (see Figure 2-4). The MBT timing varies with engine speed, load, air/fuel ratio, fuel type and some other operating conditions. In practice, it is difficult to precisely determine the MBT timing due to the flatness of the curve at which the performance is optimum, therefore the MBT timing is usually defined as the minimum spark advance that gives the best torque.

In the current study, all the performance and exhaust emission tests were carried out with the spark timing being manually adjusted to the MBT timing except for those tests that involve varying the spark timing. As mentioned before, the test engine is equipped with an electronic ignition system with an electronic timing adjustment. This system adjusts the spark timing to preset values depending on the engine speed and the intake vacuum that represents the engine load. The system has no means for mechanical adjustment of spark timing. In order to adjust the timing, the intake vacuum hose that carries the load signal to the ignition system controller was disconnected from the intake manifold and connected instead with a manual vacuum pump as shown in Figure 3-7. In other words, the load signal was controlled so that the ignition system controller adjusts the spark timing as desired. The adjustment by this method is limited between the preset minimum and maximum values at a specific speed. The adjustment range can be slightly modified by a fitted octane adapter.

3.4.2 Air/fuel Ratio

The air/fuel ratio has a significant influence on the engine performance and exhaust emissions. For higher output and a smoother operation, engines are usually operated with a rich mixture ($\Phi > 1.0$). However, the rich mixture operation is characterized by a significant increase in carbon monoxide (CO) and hydrocarbon (HC) emissions. The specific fuel consumption also increases which means a decrease in thermal efficiency. For better fuel economy and lower CO and HC emissions, engines are operated with a lean mixture ($\Phi < 1.0$). One disadvantage of the lean mixture operation is the production

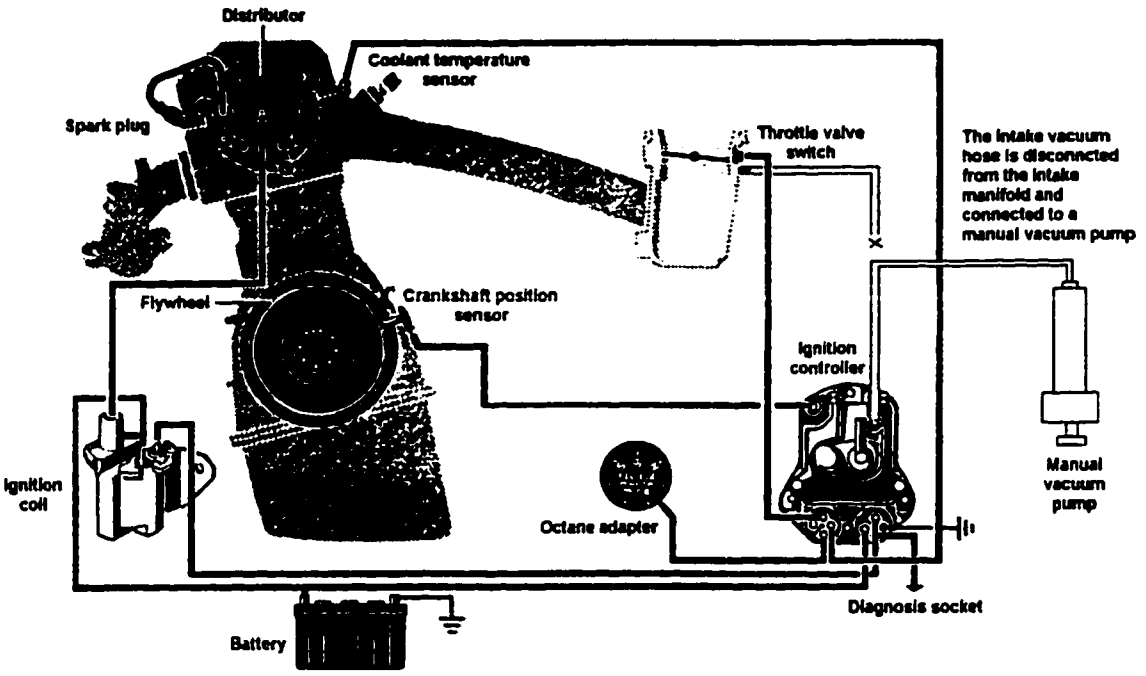

Figure 3-7: A schematic diagram for the ignition system of the test engine with the manual vacuum pump attached to the ignition controller [38].

of more nitrogen oxides (NOx). One of the advantages of replacing lead additives with oxygenates will be the possibility of utilizing the three-way catalysts and thus achieving the maximum possible reduction in the major exhaust emission species. This catalyst needs to be operated with stoichiometric mixtures in order to work efficiently. In the light of these facts, all the performance and emissions tests in the current study were conducted with the engine operating with a stoichiometric mixture except for those tests that involve variation of equivalence ratio.

The test engine used in the current study is equipped with a multi-point continuous fuel injection system called KE-Jetronic. This system contains an electrohydraulic pressure actuator capable of controlling the air/fuel ratio. To make use of this actuator and to avoid the tedious mechanical adjustment, a special device was constructed to electrically control the air/fuel ratio by supplying a small controlled direct current to the actuator

through a dc generator as shown in Figure 3-8. The mechanical adjustment was used in cases where electrical control could not achieve the desired setting.

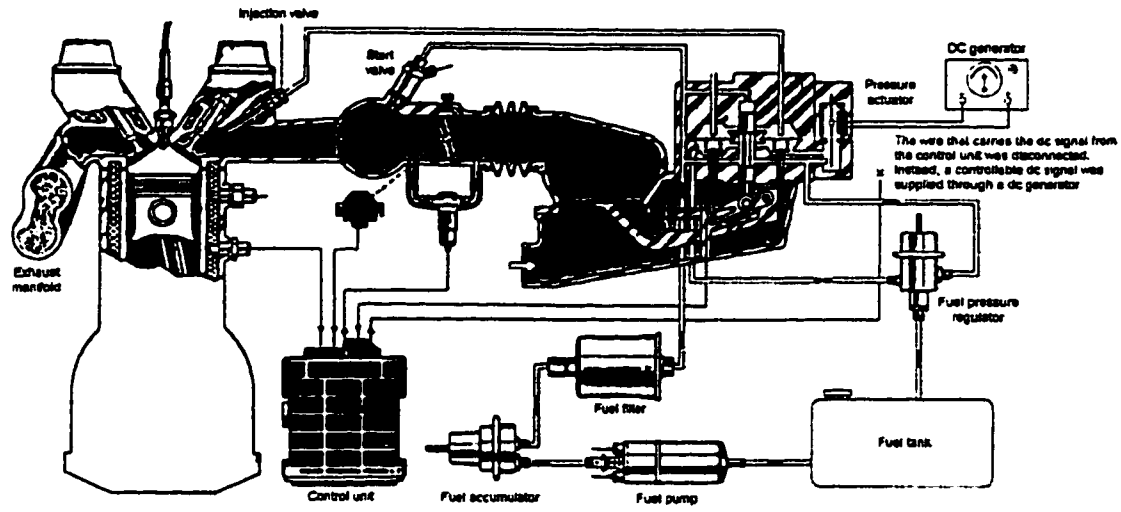

Figure 3-8: A schematic diagram for the fuel system of the test engine with the dc generator attached to the fuel-pressure actuator [39].

3.4.3 Other Operational Conditions

The temperatures of the cooling water and the lubrication oil were controlled by two fitted heat exchangers. In all the tests, the cooling water temperature was kept at 80 ±5 °C. The temperature of the lubrication oil was kept at 80 ±2 °C during the exhaust emission tests, and ranged from 80 °C to 95 °C during the performance tests. The test room temperature was kept at 25 ±2 °C during the exhaust emission tests and ranged from 20 °C to 30 °C during the performance tests. The recorded atmospheric pressure in the test room ranged from 99.4 kPa to 100.9 kPa. The engine performance results were corrected to the reference conditions (standard dry-air absolute pressure = 99 kPa, standard temperature = 25 °C) using the following relation:

$$C_F = \frac{99}{P_D}\left(\frac{T_M}{25+273}\right)^{1/2} \quad \text{(Equation 3-1)}$$

where C_F is the correction factor, P_D is the ambient dry-air pressure, and T_M is the ambient temperature.

3.4.3 Test Procedure

A variable-speed wide-open throttle test was carried out to measure the maximum torque (or maximum brake mean effective pressure, bmep) that the engine can exert and the corresponding brake thermal efficiency. Throughout the test, the spark timing was adjusted to MBT timing and the mixture to stoichiometric (Φ=1.0). The volumetric concentrations of carbon monoxide (in vol. %), the unburned hydrocarbons (in ppm C), and the nitrogen oxides (in ppm) in the exhaust gas were measured at a wide range of operating conditions. The exhaust sample was taken from the exhaust pipe just before the muffler. The controlled variables during the exhaust emission tests were the engine speed, load, ignition timing, and equivalence ratio. Extensive preliminary tests were conducted to decide the practical testing range of each variable. Certain limits have been imposed on these ranges due to either unsatisfactory engine operation or equipment limitation. The effect of speed on the tested exhaust emissions was evaluated at different engine loads. During this test, MBT timing and stoichiometric mixture were maintained. The effects of spark timing and equivalence ratio were also evaluated. These later tests were conducted at constant load and speed. The spark timing was kept at MBT timing in the case of equivalence ratio test, and the mixture was stoichiometric at the spark timing test. The exhaust emissions were also measured at the idle speed for different spark timings. In the case of idle speed test, a very small load was applied (about 10 Nm) in order to stabilize the engine operation and thus the measurements. The test procedure is outlined in Figure 3-9 along with the conditions at which the various tests were conducted. Before every test, the engine was allowed to warm up for about half an hour and the gas analyzers were calibrated with known-composition

gases. At each test, enough time was allowed for the readings to stabilize before they are recorded.

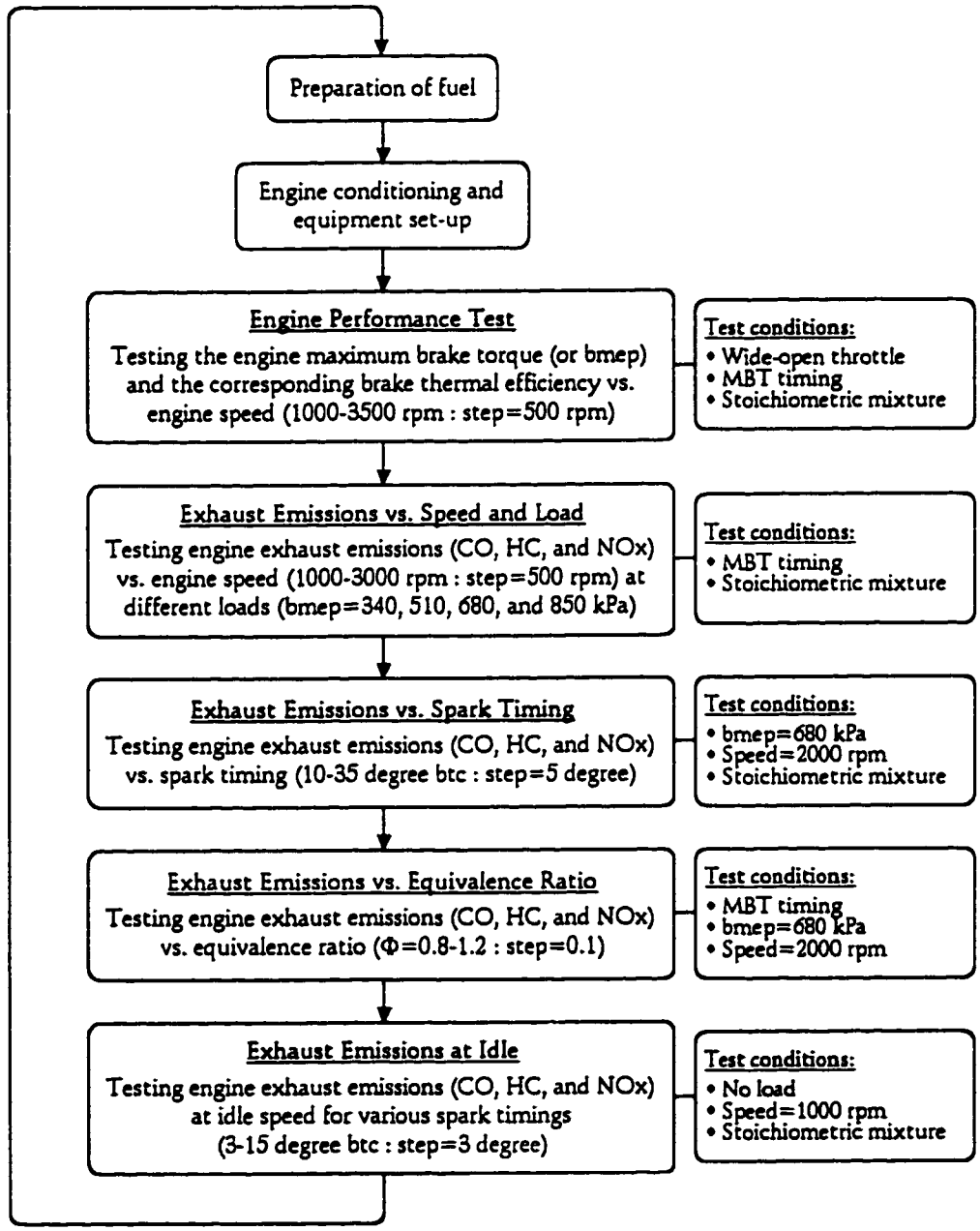

Figure 3-9: An outline of the test procedure and test conditions.

Chapter 4

Performance Results and Discussion

In this chapter, the results obtained from the performance tests will be presented and discussed. The performance was evaluated in terms of the maximum output and the brake thermal efficiency of the engine. The maximum output was measured in a wide-open throttle variable-speed test. In the same test, the corresponding values for the brake thermal efficiency were evaluated. The brake thermal efficiency was also evaluated at a constant-speed constant-load test as a function of the equivalence ratio. The MBT timing values and the exhaust gas temperatures were closely examined in order to help in understanding the performance results and explaining the variations between the fuels.

4.1 Variable Speed Test

The engine maximum output and brake thermal efficiency were evaluated as a function of engine speed. During the test, the spark timing was adjusted to maximum brake torque (MBT) timing and the mixture was set to stoichiometric. The maximum brake torque timing is the spark timing at which the engine performance is optimum. At MBT timing, the engine provides the maximum torque at constant throttle and air/fuel composition. The MBT timing is also characterized by the minimum specific fuel consumption and thus the maximum thermal efficiency [1, 2, 4].

Figures 4-1, 2, and 3 show the values of the MBT timing versus engine speed for all the tested fuels. These values represent the MBT timing at wide-open throttle and stoichiometric mixture. The MBT timing is given in terms of crankshaft degrees before the top center (btc). The results show that the MBT timing is continuously advancing as the engine speed increases. This advancing compensates for the decreasing time corresponding to a crankshaft degree due to the increasing engine speed. The base fuel

showed the least advanced MBT timing. This is due to the fact that at high engine loads, the MBT timing is limited by the occurrence of knock, which is dependent on the anti-knock property of the fuel [4]. As the fuel becomes more resistive to knock, the ignition spark can be set to occur earlier in the compression stroke. Since the base fuel has the least resistivity to knock, as indicated by its lowest octane number among the tested fuels (Table 3-2), its knock limit will be closer to the top center resulting in less advanced MBT timing.

The addition of all oxygenates, which have much higher octane number than gasoline, increased the octane number of the blends and resulted in more advanced MBT timing than the base fuel. The highest increase occurred in the case of ethanol blends while the lowest occurred in the case of MTBE blends. Although the octane number is usually used to indicate the fuel resistivity to knock, it only reflects the fuel anti-knock behavior at certain operating conditions. It does not necessarily form a good basis for comparison between the anti-knock behavior of different fuels at all operating conditions. For this reason, not all the MBT timing results can be explained by the octane differences.

Figures 4-4, 5, and 6 show the variation of exhaust gas temperature corresponding to the current test for all the tested fuels. These exhaust temperatures were measured near the outlet of the exhaust manifold. In general, the highest exhaust temperature is observed with the base fuel, and the lowest with the leaded fuel. In addition, the exhaust temperature decreases as the oxygenate ratio in the blend increases. These variations in exhaust temperatures can be attributed to the increase in thermal efficiency and/or the decrease in the combustion temperatures. The increase in thermal efficiency due to the more advanced spark timing means that a larger portion of combustion heat has been converted into work and therefore lower exhaust temperatures are expected. In addition, the lower combustion temperatures characterizing the oxygenated blends are expected to result in lower exhaust temperatures [2, 8, 11].

4.1.1 Maximum Engine Output

The engine maximum output was measured in terms of the maximum brake torque exerted by the engine at different engine speeds. The brake mean effective pressure (bmep) is usually used instead of the brake torque to represent the engine output. The bmep is a more universal term than the brake torque since it is independent of the engine size (refer to Equation 1-3). For this reason, the engine output results are presented here in terms of brake torque along with the corresponding bmep values.

The maximum brake torque and bmep results versus engine speed for all the tested fuels are shown in Figures 4-7, 8, and 9. The points indicate the measured values while the lines indicate the least-squares polynomial fit. Consistent and persisting fluctuations in the maximum torque measurements were observed. These fluctuations were closely examined and eventually linked to abnormal vibrations in the test set-up. A possible cause of these abnormal vibrations is a misalignment of the shaft connecting the engine with the dynamometer. The vibrations of the test set-up were measured in terms of the average noise level close to the engine. Figure 4-10 shows the variation of the average noise level along with the maximum torque at wide-open throttle test.

Generally, the brake torque (and bmep) increased sharply with increasing engine speed reaching its maximum at about 2500 rpm. From 2500 to 3500 rpm, the brake torque either remained constant or slightly decreased. The drop in brake torque at lower engine speeds is due mainly to the lower volumetric efficiency and the significance of heat losses to the cylinder walls. The peaking of the brake torque at mid-range speed, about 2500 rpm in this case, is a result of the optimum volumetric efficiency and minimum power losses normally attained at this speed range. At higher speeds, the decline in volumetric efficiency and the sharp increase in friction losses cause the brake torque to decrease [2].

The base fuel produced the lowest brake torque among all the tested fuels. The leaded fuel exhibited a substantial increase in the brake torque with respect to the base fuel. This substantial increase is a result of the improved anti-knock behavior due to the

addition of TEL, which raised the octane number from 84.7 for the base fuel to 92 for the leaded fuel. The improved anti-knock behavior allowed a more advanced MBT timing that results in higher combustion pressure and thus higher exerted torque (and bmep) [2, 4].

Generally, the results of MTBE blends (Figure 4-7) indicate an increase in brake torque with respect to the base fuel. The significance of this increase varied with engine speed and MTBE ratio in the blend. At lower speeds, increasing MTBE ratio in the blend resulted in a gradually slight increase in the brake torque. At higher speeds, however, a considerable increase in the brake torque was obtained with the blend containing 10 vol. % of MTBE (MTBE10), but further addition of MTBE eventually led to the decline of brake torque. The gain in brake torque obtained with MTBE blends can be attributed to the improvement in anti-knock behavior, which allows more advanced MBT timing and thus higher output [2, 4]. As the MTBE ratio in the blend increases, however, the decreasing heating value tends to offset the improvement in performance that results from better anti-knock behavior. The positive effect of MTBE on maximum brake power has been reported also by Wang et al. [24].

The results for the methanol blends (Figure 4-8) indicate an increasingly improving brake torque with the increasing methanol ratio in the blend. The improvement in the brake torque persists over the whole range of the tested engine speed. This gain in brake torque obtained with methanol blends can be attributed to the better anti-knock behavior of these blends and the improvement in engine volumetric efficiency. The research octane numbers (RON) for the methanol blends are significantly higher than that of the base fuel (Table 3-2). This improved anti-knock behavior allows a more advanced MBT timing, as evident in Figure 4-2, and thus a higher engine output. The improvement in engine volumetric efficiency is a result of the higher latent heat of vaporization characterizing the methanol blends. The latent heat of vaporization is the amount of heat required to vaporize the liquid molecules of the fuel prior to the

combustion process. Most of this heat, especially in the case of port fuel injection systems, is absorbed from the air accompanying the fuel in its way to the engine cylinder. The absorption of heat from the air cools it and makes it denser. This allows more air mass to be admitted into the cylinder during the induction process and thus increasing the volumetric efficiency. Alcohols in general are characterized by their higher latent heat of vaporization when compared to the typical gasoline. Increasing alcohol ratio in a gasoline-alcohol blend will increase the latent heat of vaporization of the blend and thus improve the engine volumetric efficiency. Alcohols, on the other hand, have lower heating values when compared to gasoline, which means a lower energy release during combustion and a lower work transfer during expansion process. However, the gain in brake torque due to the improvement in anti-knock behavior and the volumetric efficiency seems to overweigh the losses due to the lower heating value [4, 8, 11, 12, 14]. The gain in maximum brake power due to the addition of methanol was reported also by Tsao and Lee [23].

The results for the ethanol blends (Figure 4-9) show that there is a significant improvement in brake torque with the 10 vol. % ethanol blend (ETH10) when compared to the base fuel. At low engine speeds, further increase of ethanol ratio had no effect on the brake torque. At high speeds, the 15 vol. % ethanol blend (ETH15) performed slightly better but further addition of ethanol resulted in a decline in the brake torque. Similar to the methanol blends, the gain in brake torque can be attributed to the improvement in anti-knock behavior and volumetric efficiency. However, the improvements seem to be significant only with low ethanol ratios (up to 15 vol. % in this case) and the decreasing of heating value resulting from increasing ethanol ratio in the blend quickly offsets these improvements. In general, the three ethanol blends resulted in higher brake torque than the base fuel. This result does not agree with the results reported by Rajan [22]. Rajan reported a sharp decrease in engine output in the case of 20 vol. % hydrated ethanol (about 6% water in ethanol).

The results for the best performing blends in terms of the maximum brake torque compared to the base and leaded fuels are shown in Figure 4-11. The performance of METH20 and ETH15 is comparable with that of leaded fuel and represents a gain of about 5% in the brake torque when compared to the base fuel. The best performing MTBE blend, which is the MTBE15, shows a gain of only about 2%.

4.1.2 Brake Thermal Efficiency

The brake thermal efficiency is an empirically defined term that indicates the effectiveness of the engine in converting the fuel energy to useful mechanical work. It is defined as the ratio between the engine brake power and the rate of fuel energy input (Equation 1-1).

Figures 4-12, 13, and 14 show the brake thermal efficiency results for the variable-speed test at wide-open throttle, stoichiometric mixture, and MBT timing. In general, the brake thermal efficiency improves with increasing speed up to about 2500 rpm where it becomes maximum then starts to decline as the speed increases. Among all the tested fuels, the base fuel obtained the lowest brake thermal efficiency along the whole range of the test speed. The results for the leaded fuel show a significant improvement in the brake thermal efficiency at lower speeds with respect to the base fuel (about 10.7%). As the speed increases, however, this improvement continues to decline and eventually vanishes. This indicates that the effect of improved anti-knock behavior of the leaded fuel on the thermal efficiency lessens as the engine speed increases.

The results for the MTBE blends, shown in Figure 4-12, indicate a significant improvement in brake thermal efficiency. As the MTBE ratio in the blend increases, the brake thermal efficiency continues to improve achieving a maximum gain of about 12.5% with respect to the base fuel at low speeds (1000 rpm). The improvement in the brake thermal efficiency with the three MTBE blends persists over the whole range of test speed. This improvement can be attributed to the more advanced MBT timing allowed by the improved anti-knock behavior, and the lower heat losses due to the lower

combustion temperatures. The MTBE blends have lower heating values than that of the base fuel and, therefore, their combustion temperatures are expected to be also lower [2, 8]. The improvement in thermal efficiency observed here agrees qualitatively with the results reported by Wang et al. [24].

The results for the methanol blends, Figure 4-13, show a continuous improvement in brake thermal efficiency as the methanol ratio in the blend increases. The improvement is maintained over the entire speed range and is maximum (about 12.5%) at mid-range speed (about 2500 rpm). The improvement in brake thermal efficiency associated with the methanol blends is also due to the more advanced MBT timing and the lower heat losses. The heating values for the methanol blends are noticeably less than that of the base fuel due to the extremely low heating value of methanol, which is less than 50% of that of typical gasoline [27]. The improvement in brake thermal efficiency due to the addition of methanol was also reported by Tsao and Lee [23] and Kowalewicsz [27].

The results for the three ethanol blends, Figure 4-14, also show an improvement in the brake thermal efficiency. A maximum gain of about 9% is achieved at mid-range speeds with the ETH20 blend. The improvement in brake thermal efficiency noticed here can be also explained by the improvement in anti-knock behavior that allows a more advanced MBT timing, and the expected lower heat losses due the lower combustion temperatures. Rajan [22] reported an improvement in the thermal efficiency with blends containing up to 10 vol. % hydrated ethanol (about 4% water in ethanol). Beyond this limit, however, Rajan reported a substantial decrease in the thermal efficiency. This contradicts the results in the current study which indicate a noticeable improvement in thermal efficiency in the case of 20 vol. % ethanol blend (which contains about 7% of water in ethanol).

Figure 4-15 shows the results of brake thermal efficiency for the best performing blends compared to the base and leaded fuels. The highest efficiency was obtained with METH20 blend followed by MTBE20 then ETH20. At low speeds, the improvement

associated with the three blends is comparable to that of leaded fuel. As the speed increases, however, the brake thermal efficiency continues to improve in the case the three blends while declining in the case of leaded fuel.

4.2 Variable Equivalence Ratio Test

The effect of equivalence ratio on the brake thermal efficiency was evaluated at a constant speed (2000 rpm), constant load (680 kPa), and MBT timing. The values for MBT timing corresponding to this test are shown in Figures 4-16, 17, and 18. In general, the MBT timing retards towards the top center as the mixture is enriched. This can be attributed to the effect of equivalence ratio on the burning rate and therefore on the time required to complete the combustion. Usually, the maximum burning rate occurs at about 10% rich mixture (i.e. Φ=1.1). Lean mixtures and extremely rich mixtures have low burning rates, and therefore, they need more advanced spark timing to attain the maximum brake torque [2]. The results show also that the MBT timing advances as the oxygenate ratio in the blend increases. This is probably due to the improvement in anti-knock behavior of the blends due the addition of oxygenates.

The exhaust gas temperatures, shown in Figures 4-19, 20, and 21, peak at about stoichiometric (Φ=1.0) and sharply drop with lean and rich mixtures. This indicates that the maximum heat loss through the exhaust process occurs in the case of stoichiometric mixtures [2]. It indicates also that the prevailing gas temperature at the end of the expansion process is higher in the case of stoichiometric than in lean or rich mixtures. In general, the exhaust temperatures for the oxygenated blends were comparable to those of the base fuel in the case of rich mixtures but noticeably lower in the case of lean mixtures. The leaded fuel exhibited a consistent decrease in exhaust temperature with respect to the base fuel over the entire test range. The decreased exhaust temperature is a result of lower combustion temperature and/or improved thermal efficiency.

Figures 4-22, 23, and 24 show the brake thermal efficiency at different equivalence ratios for all the test fuels. The results indicate that the brake thermal efficiency is significantly influenced by the equivalence ratio. The efficiency drastically deteriorates with increasing richness of the mixture ($\Phi>1.0$). On the other hand, the efficiency improves as the mixture is leaned out up to about $\Phi=0.9$ after which the improvement slows down. The deterioration of efficiency with rich mixtures results from incomplete combustion of fuel due to the lack of enough oxygen. The best combustion quality, and thus best fuel conversion efficiency, is normally attained with 10% lean mixtures that insure maximum utilization of the supplied fuel. Additional leaning of the mixture will harm the combustion quality and increase the possibility of misfiring [2, 4].

Among all the tested fuels, the base fuel attained the lowest brake thermal efficiency values in the whole test range. In general, the addition of oxygenates resulted in a noticeable improvement in the brake thermal efficiency. The results here indicate, however, less significant differences between the test fuels than in the case of wide-open throttle tests. This can be attributed to the fact that the thermal efficiency improves noticeably as the load increases because of decreasing significance of heat losses [2].

The results for the MTBE blends, shown in Figure 4-22, indicate a continuous improvement in the efficiency as the MTBE ratio increases. This improvement is sustained over the entire tested range of equivalence ratio. The maximum gain in efficiency with respect to the base fuel is observed at the rich side ($\Phi=1.2$) and is about 6%. The gain in efficiency decreases as the mixture is leaned out reaching about 3.8% at $\Phi=0.8$. In the case of the methanol blends, the efficiency also continues to improve as the methanol ratio increases in the blend at all equivalence ratios as shown in Figure 4-23. However, the gain in efficiency in this case is almost constant at all equivalence ratios and is about 6%. Different from the MTBE and methanol blends, the ethanol blends at rich equivalence ratios ($\Phi>1.0$) show a slight gain in efficiency (about 1.5%) with respect to the base fuel (Figure 4-24). Furthermore, this slight gain in efficiency is

not affected by the increase of ethanol ratio in the blend. As the mixture is leaned out, however, the gain in efficiency increases and the variation between the three ethanol blends becomes more noticeable. A maximum gain in efficiency of about 6% is observed for ETH20 at Φ=0.8.

Similar to the previous test, the improvement in efficiency noticed in this test can be attributed to the more advanced MBT timing allowed by the improved anti-knock behavior, and/or the lower heat losses due to the lower combustion temperatures.

The brake thermal efficiency values as a function of equivalence ratio for the best performing blends compared to the base and leaded fuels are shown in Figure 4-25. Again, the METH20 is the best performer at the whole test range. At the rich side (Φ>1.0), the results for the MTBE blends are comparable to those of METH20. At the lean side (Φ<1.0), the METH20 outperforms all other test fuels and is approached only by the ETH20 blend at Φ=0.8. The performance of the leaded fuel at rich equivalence ratios is comparable to the best performing blend, but fall behind at the lean side.

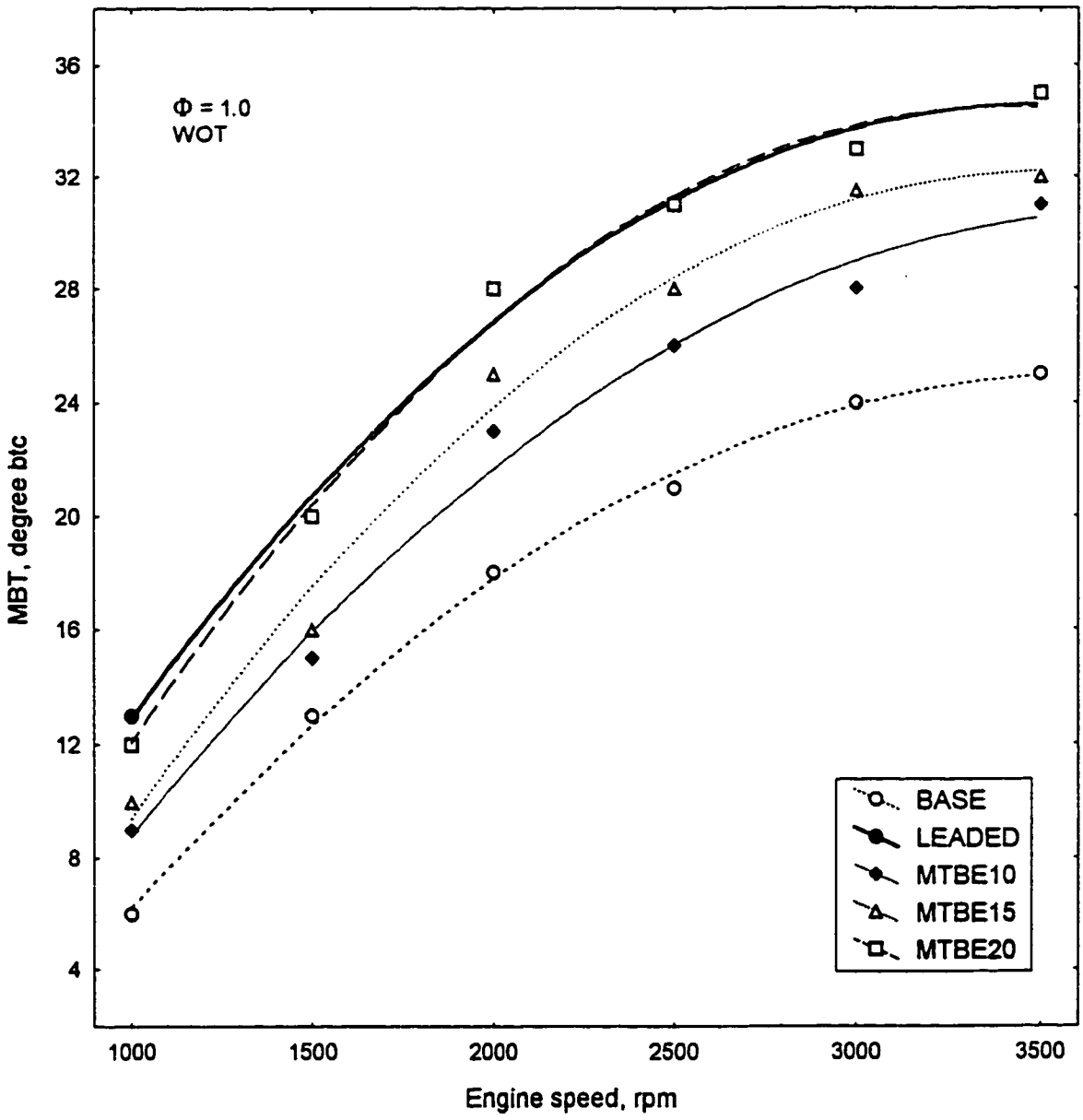

Figure 4-1: MBT spark timing at wide-open throttle for the MTBE blends.

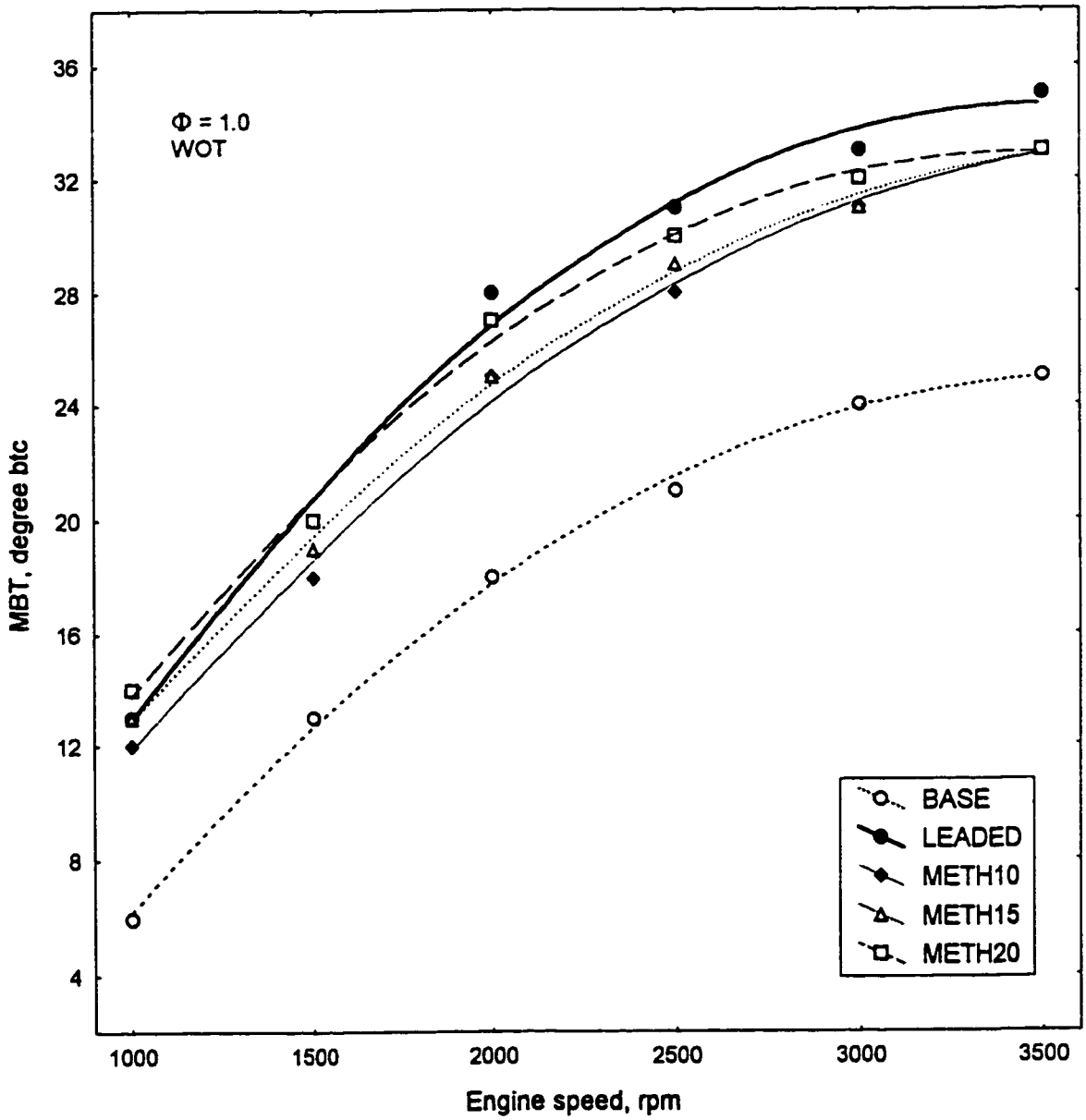

Figure 4-2: MBT spark timing at wide-open throttle for the methanol blends.

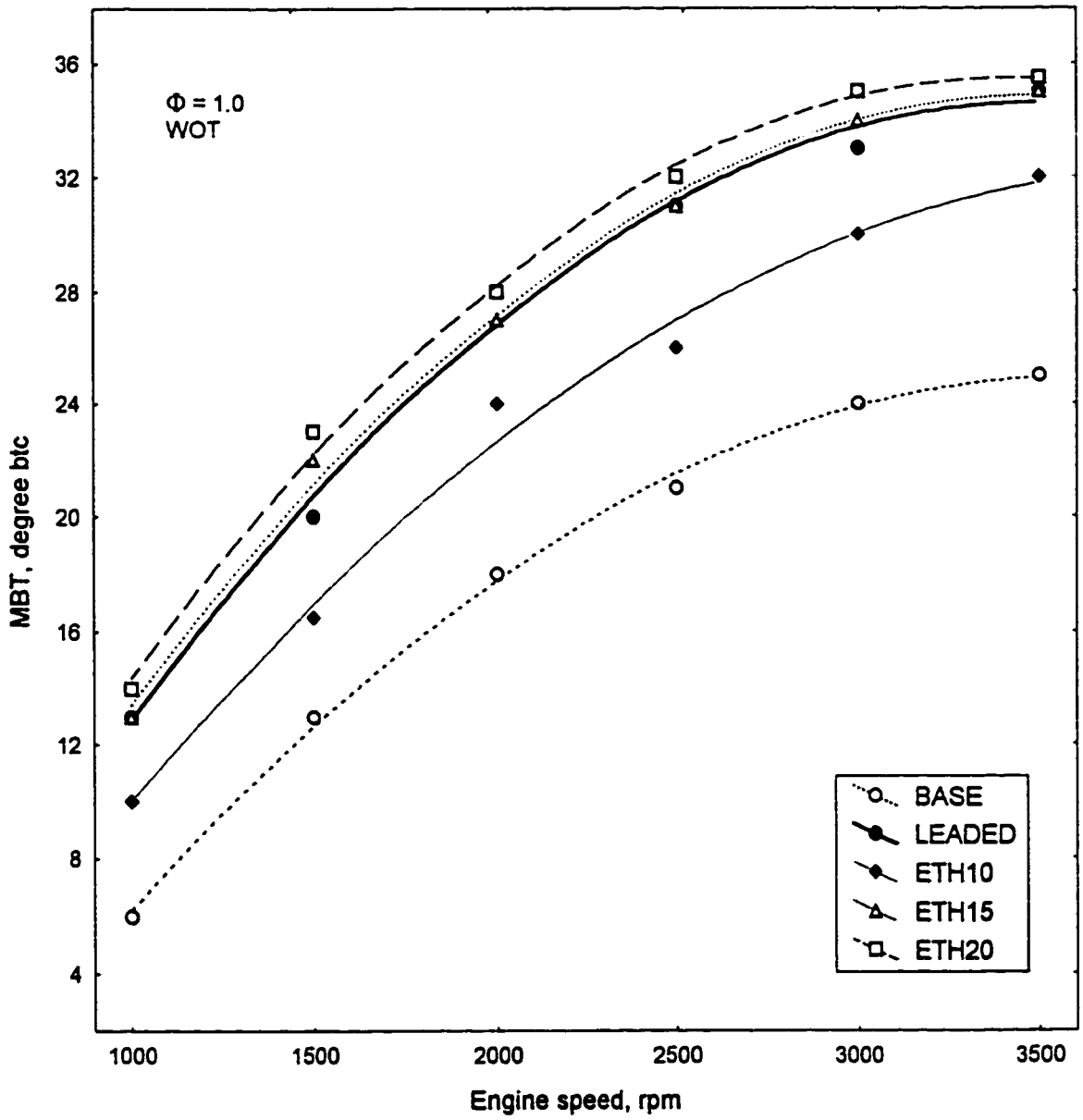

Figure 4-3: MBT spark timing at wide-open throttle for the ethanol blends.

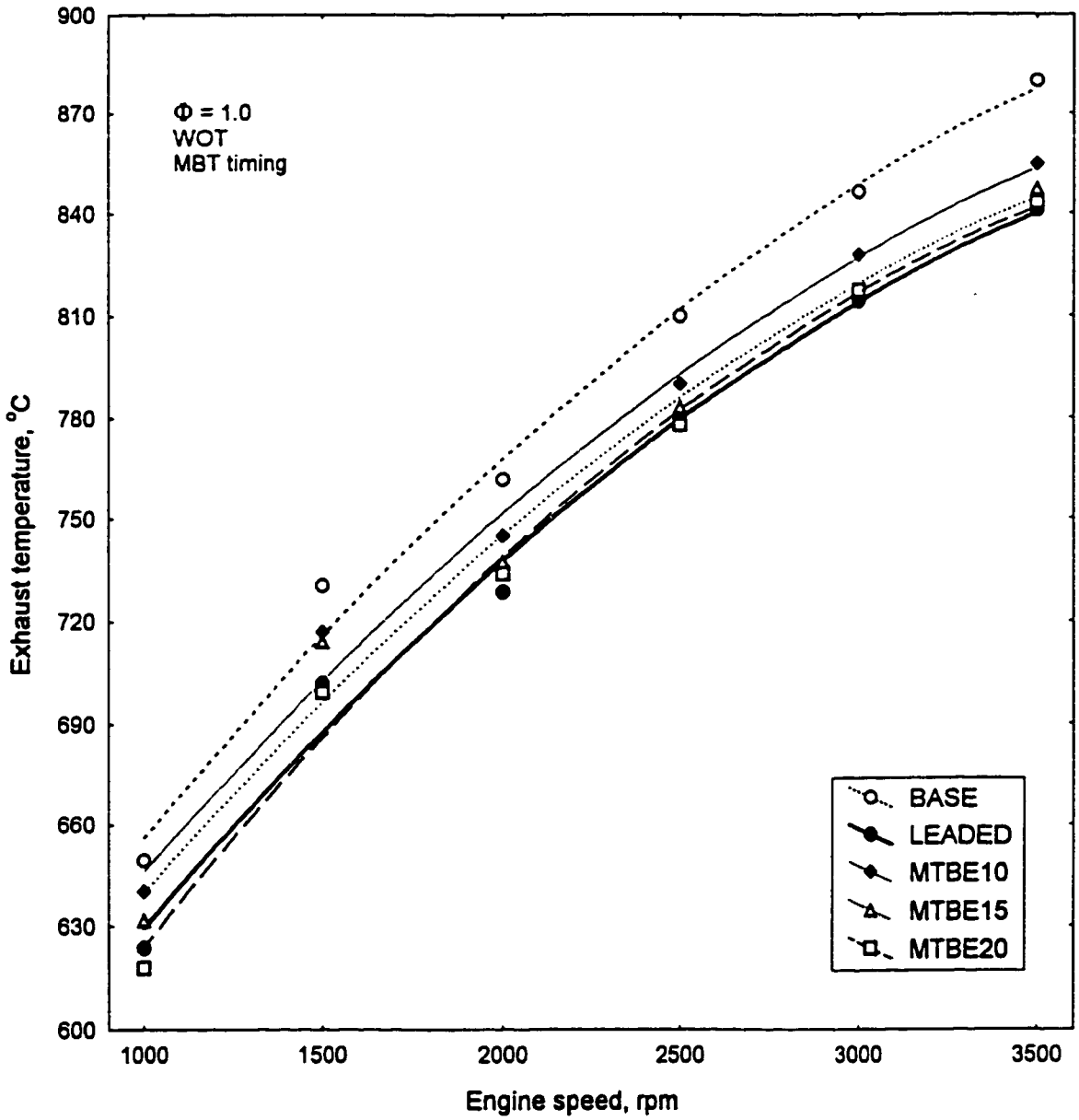

Figure 4-4: Exhaust gas temperatures at wide-open throttle for the MTBE blends.

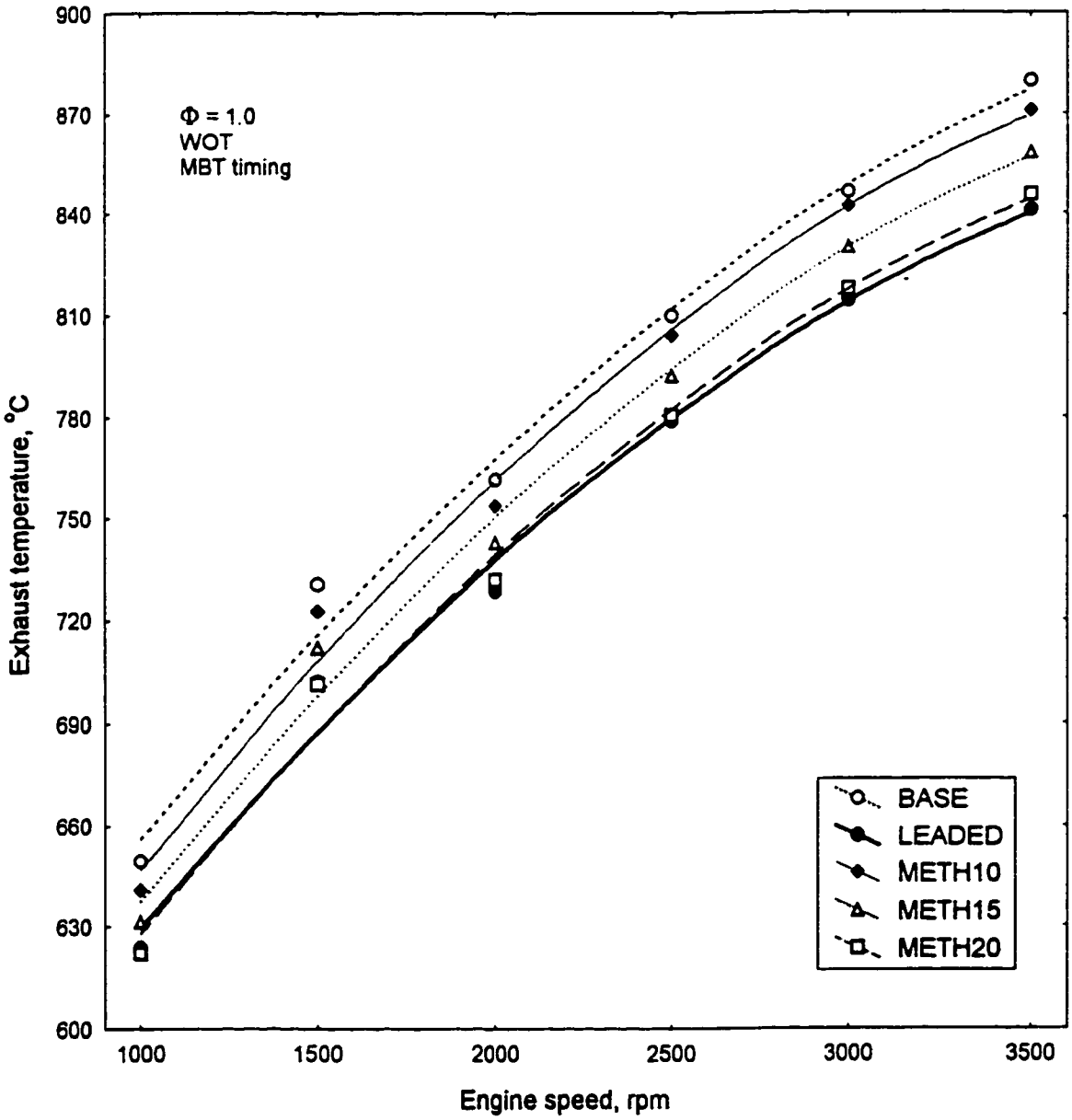

Figure 4-5: Exhaust gas temperature at wide-open throttle for the methanol blends.

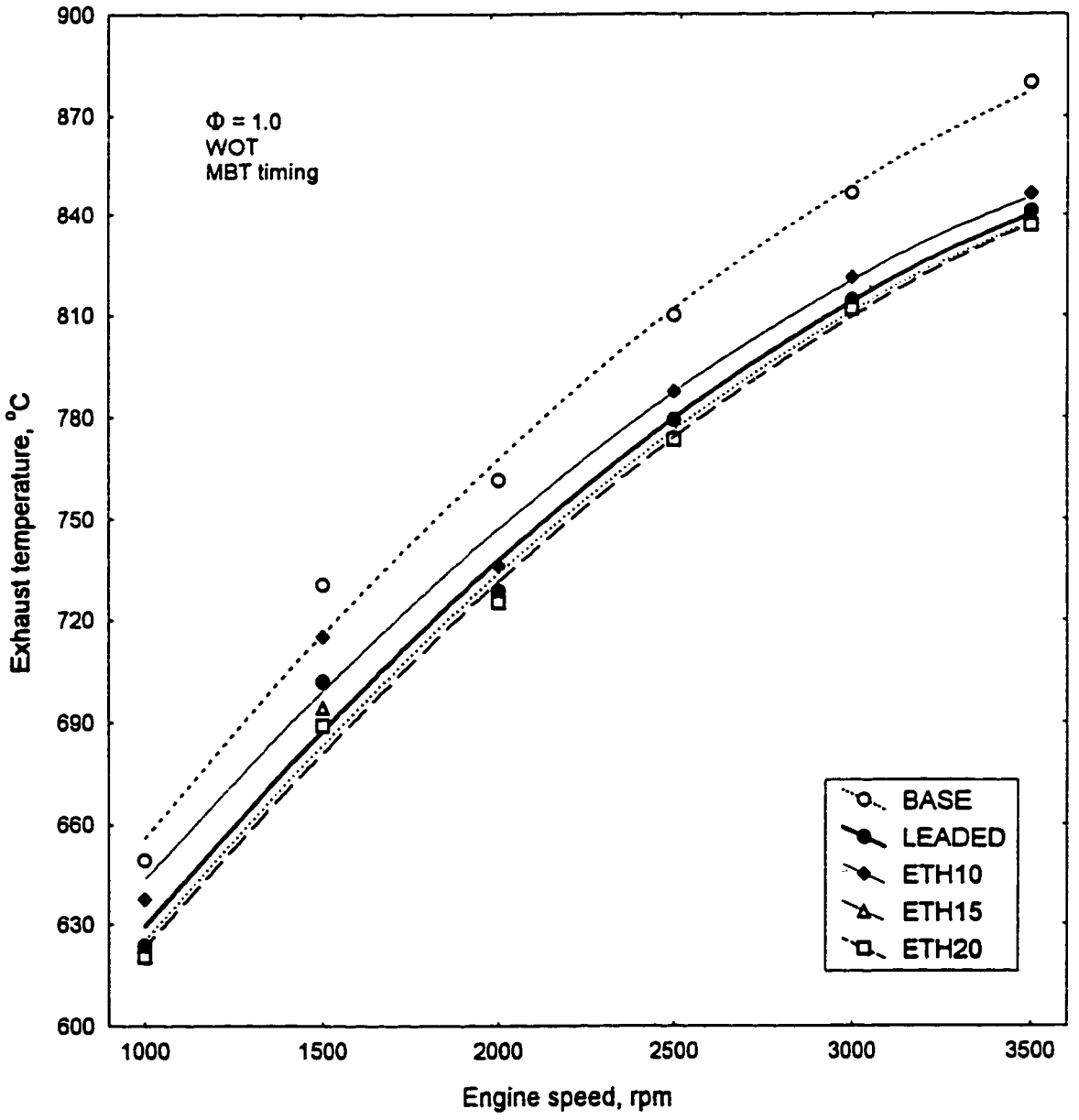

Figure 4-6: Exhaust gas temperature at wide-open throttle for the ethanol blends.

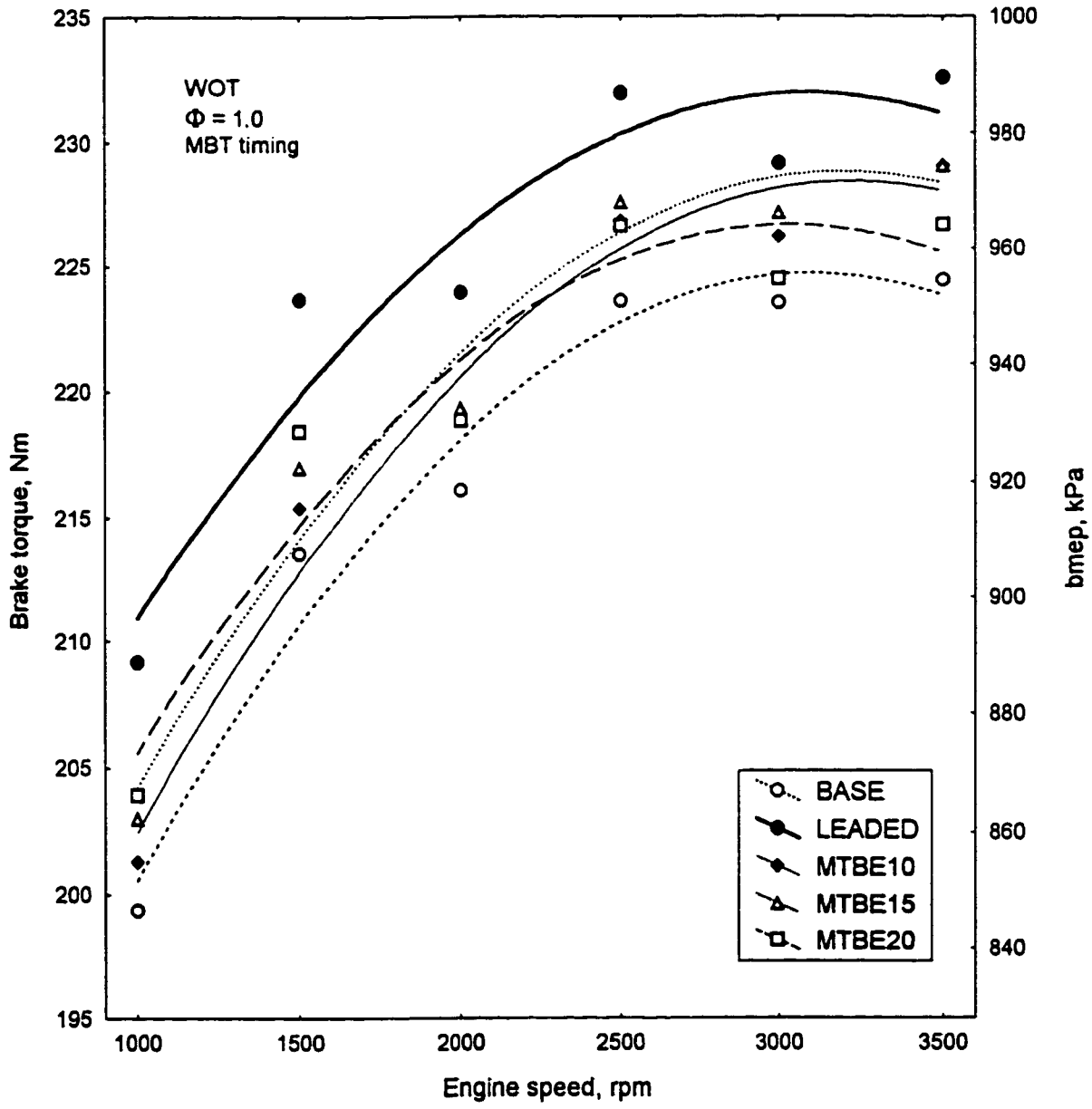

Figure 4-7: Brake torque and mean effective pressure at wide-open throttle for the MTBE blends.

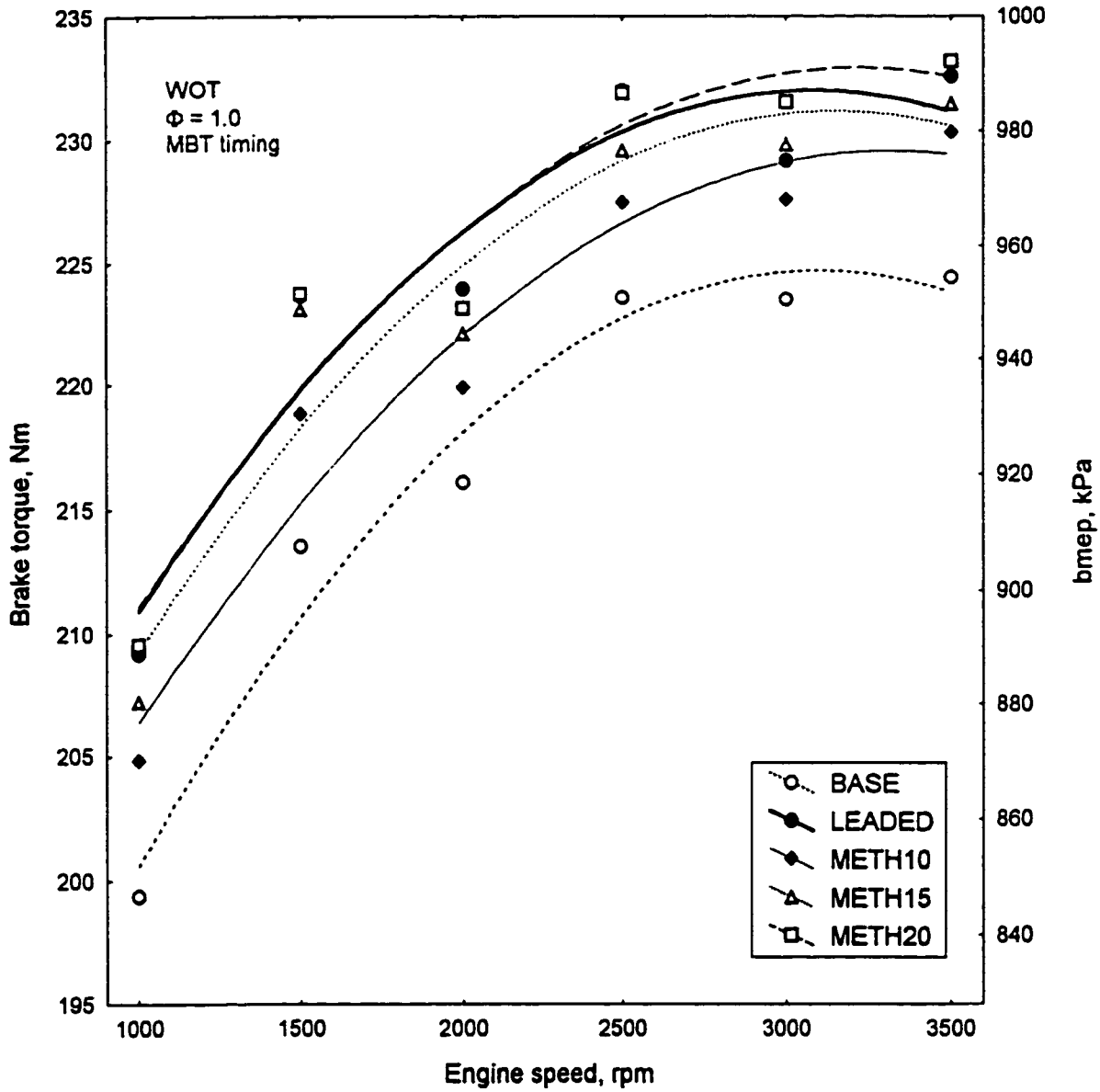

Figure 4-8: Brake torque and mean effective pressure at wide-open throttle for the methanol blends.

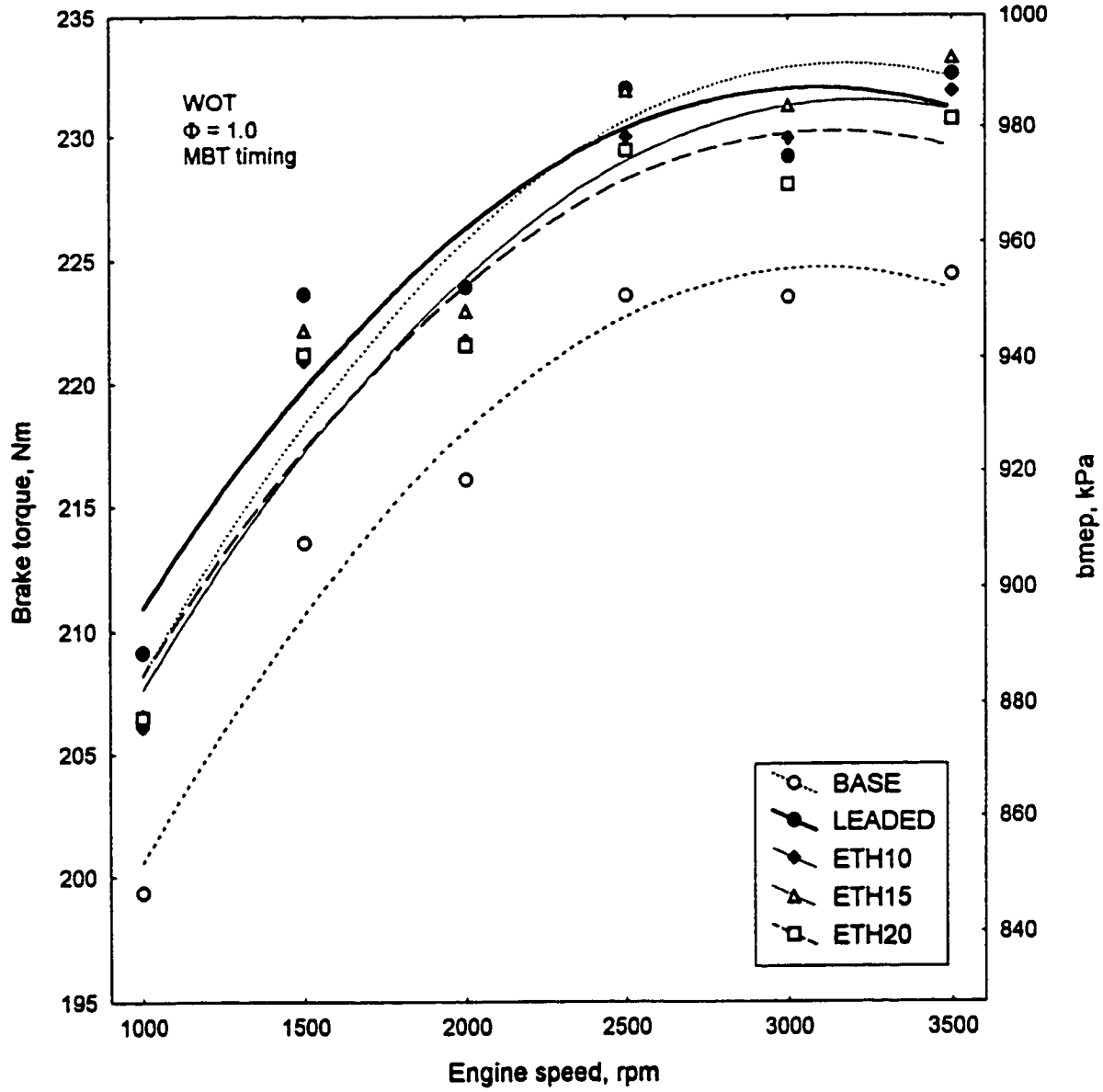

Figure 4-9: Brake torque and mean effective pressure at wide-open throttle for the ethanol blends.

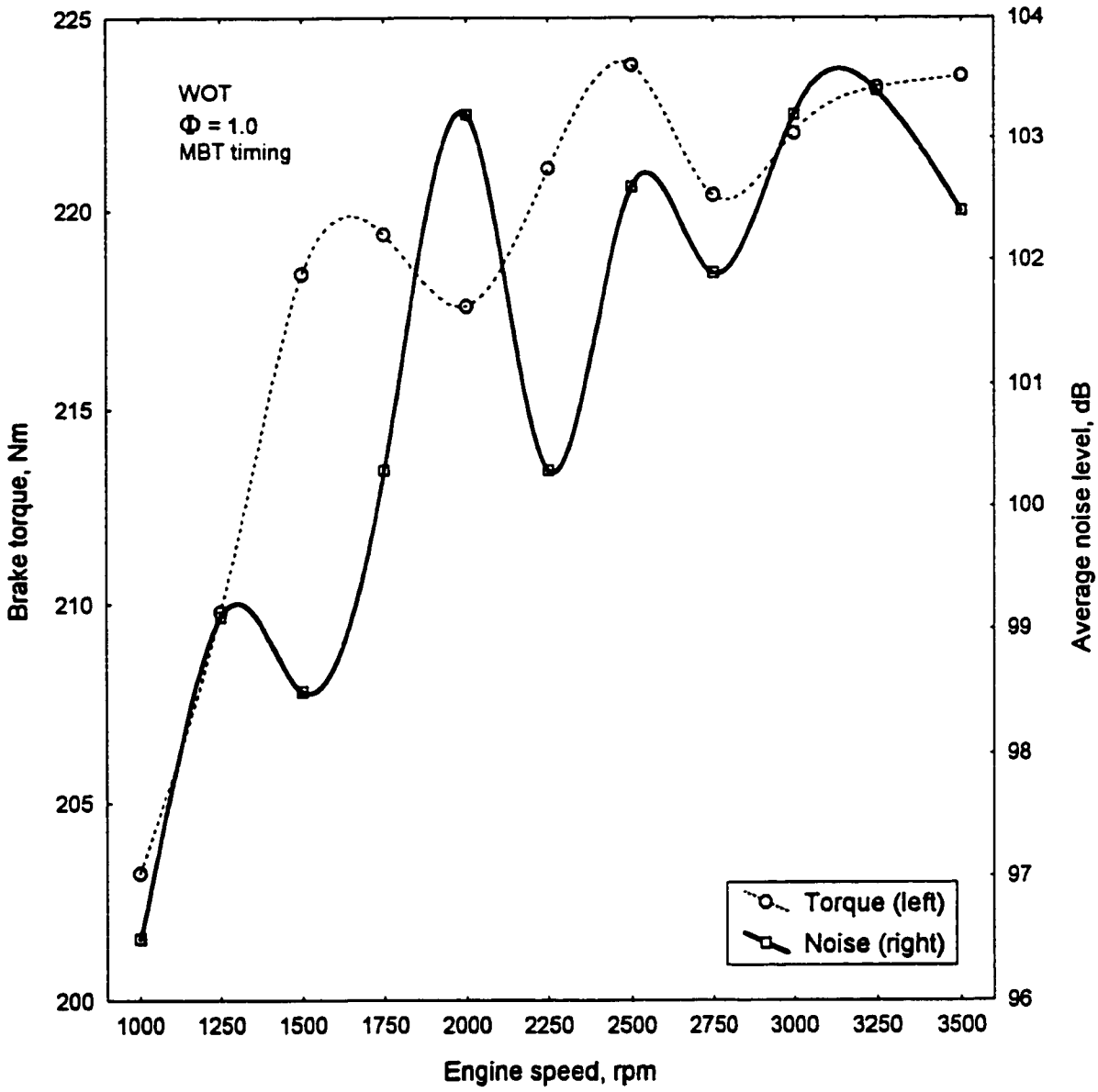

Figure 4-10: The fluctuations in brake torque and engine noise level during a normal wide-open-throttle variable-speed test.

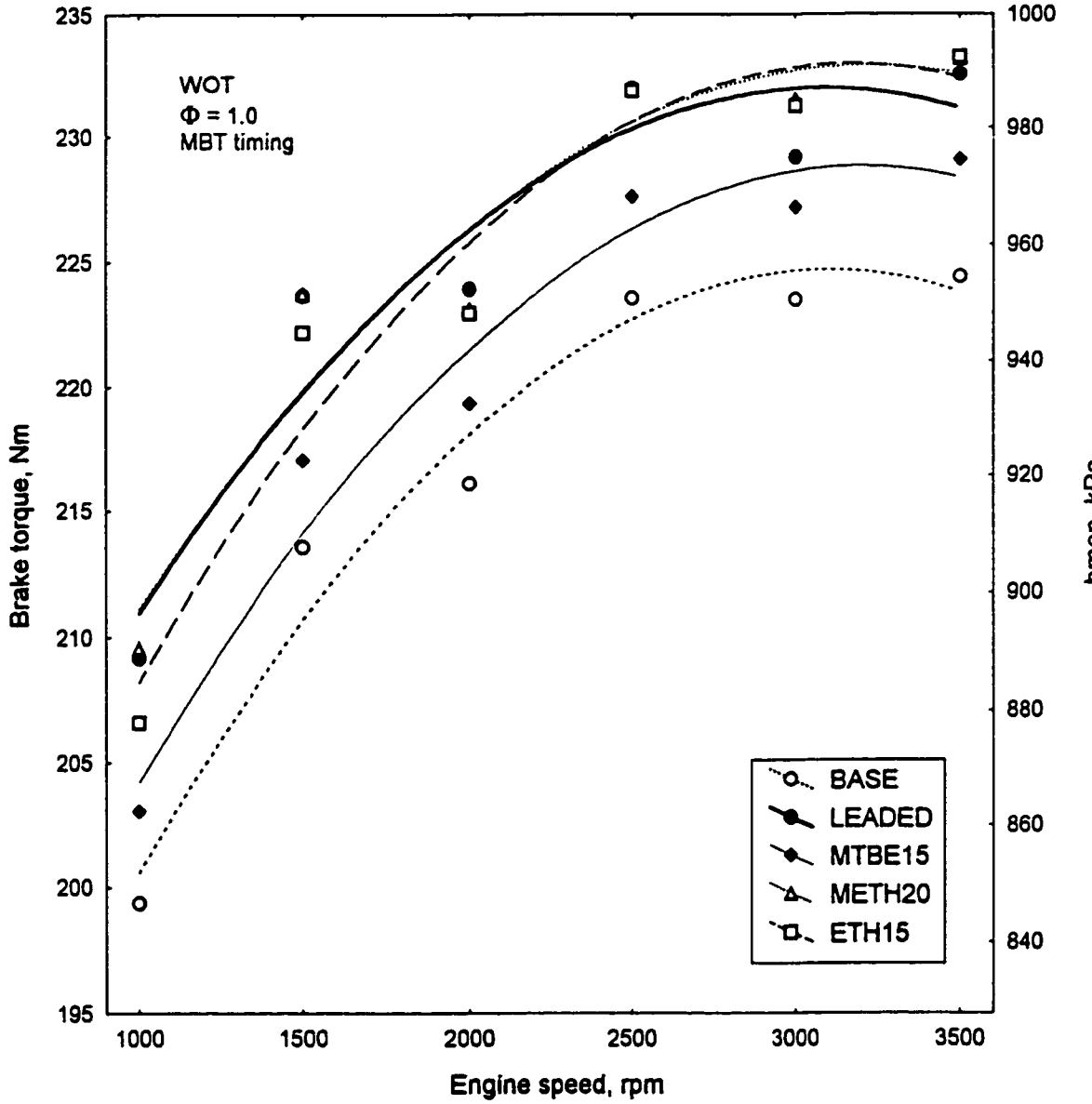

Figure 4-11: Brake torque and mean effective pressure at wide-open throttle for the best performing blends compared to the base and leaded fuels.

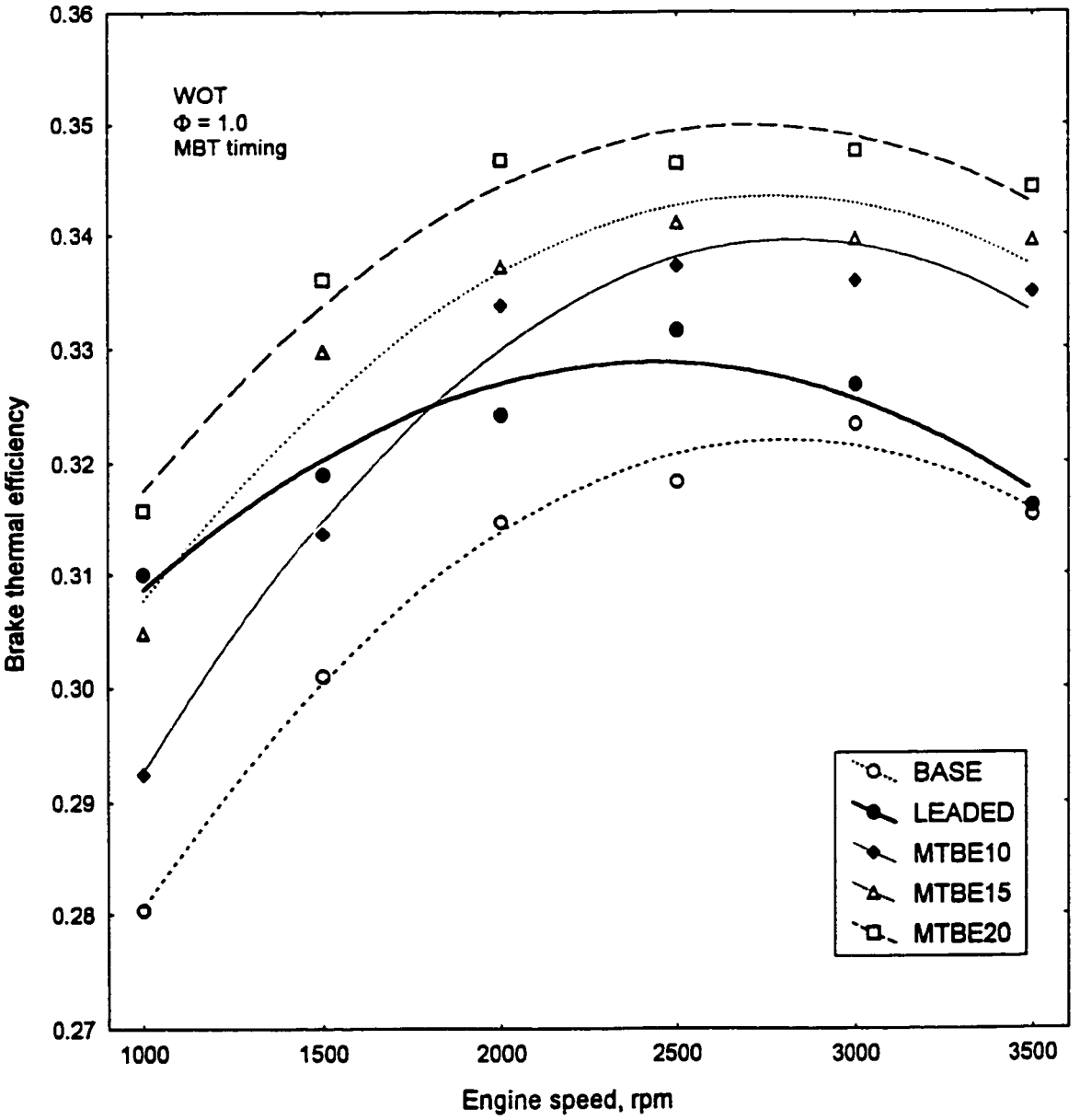

Figure 4-12: Brake thermal efficiency at wide-open throttle for the MTBE blends.

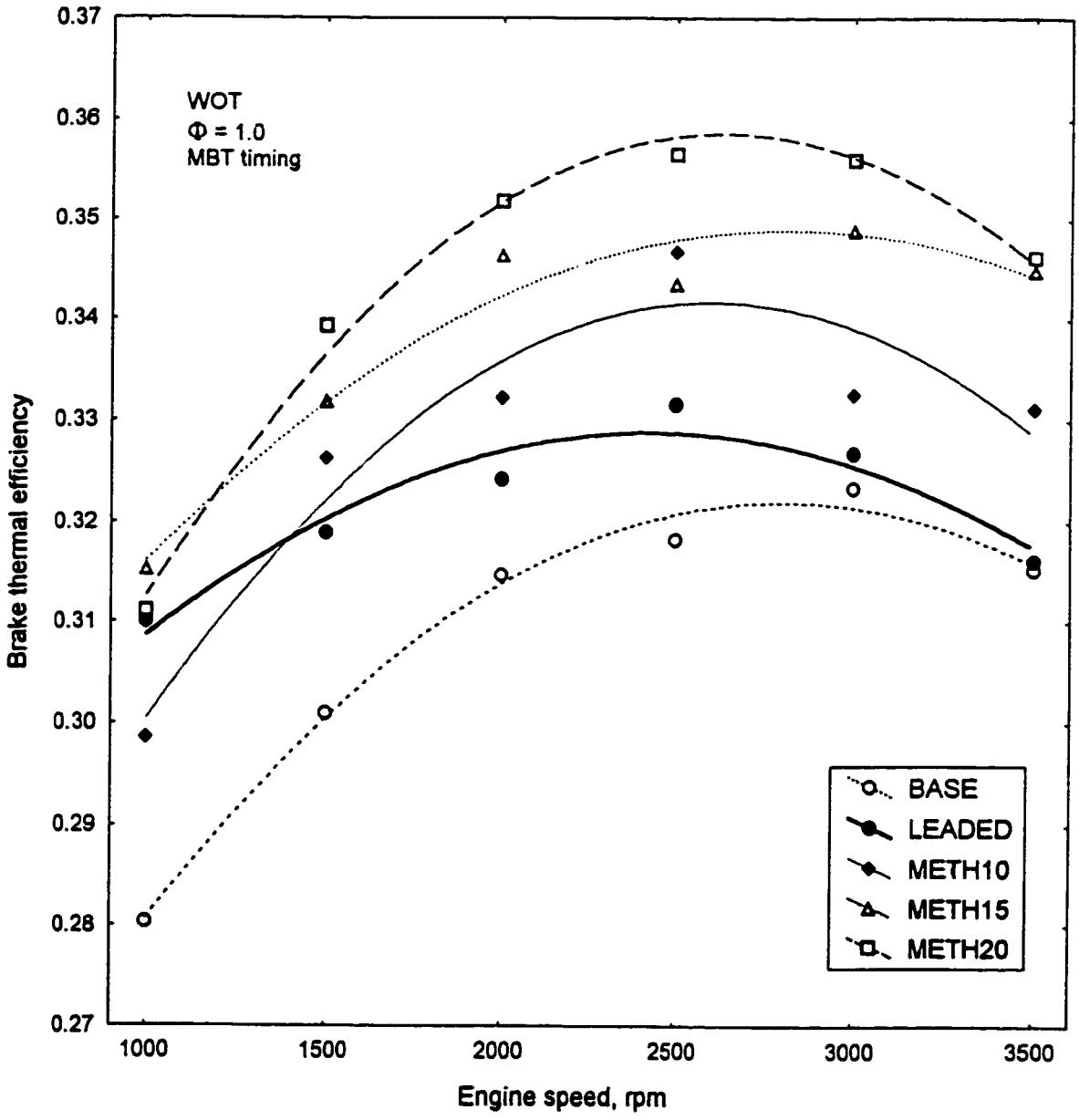

Figure 4-13: Brake thermal efficiency at wide-open throttle for the methanol blends.

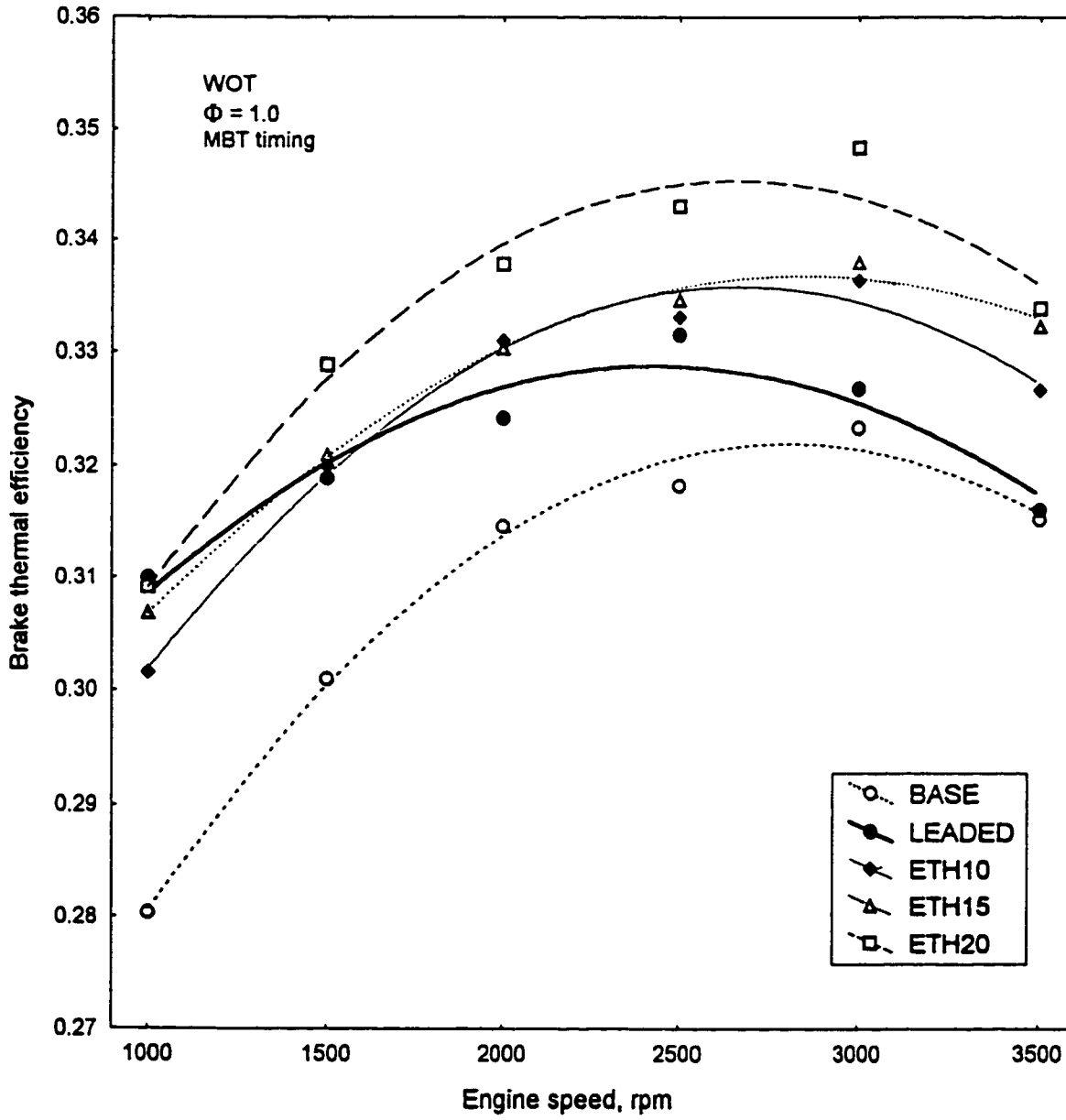

Figure 4-14: Brake thermal efficiency at wide-open throttle for the ethanol blends.

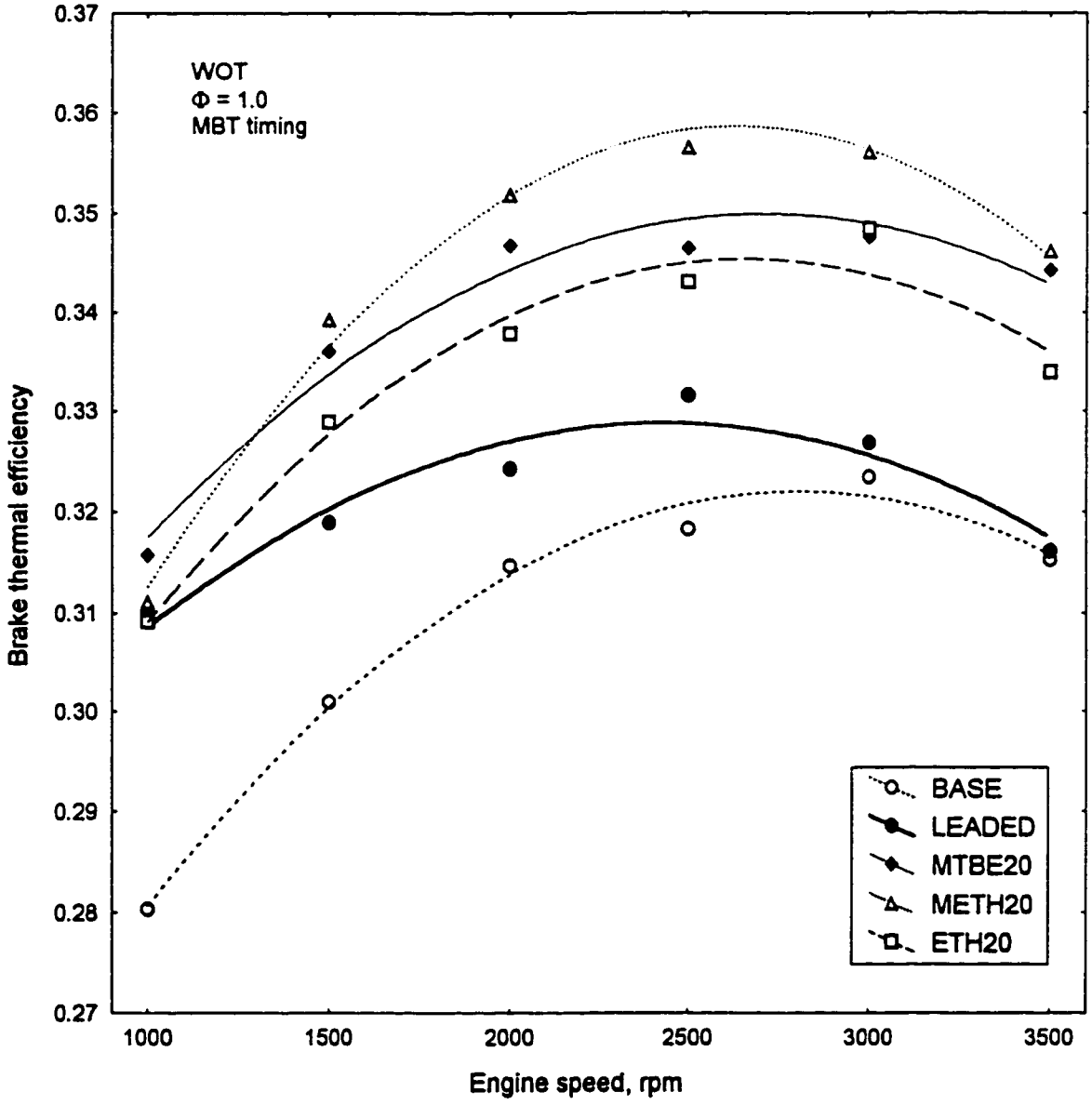

Figure 4-15: Brake thermal efficiency at wide-open throttle for the best performing blends compared to the base and leaded fuels.

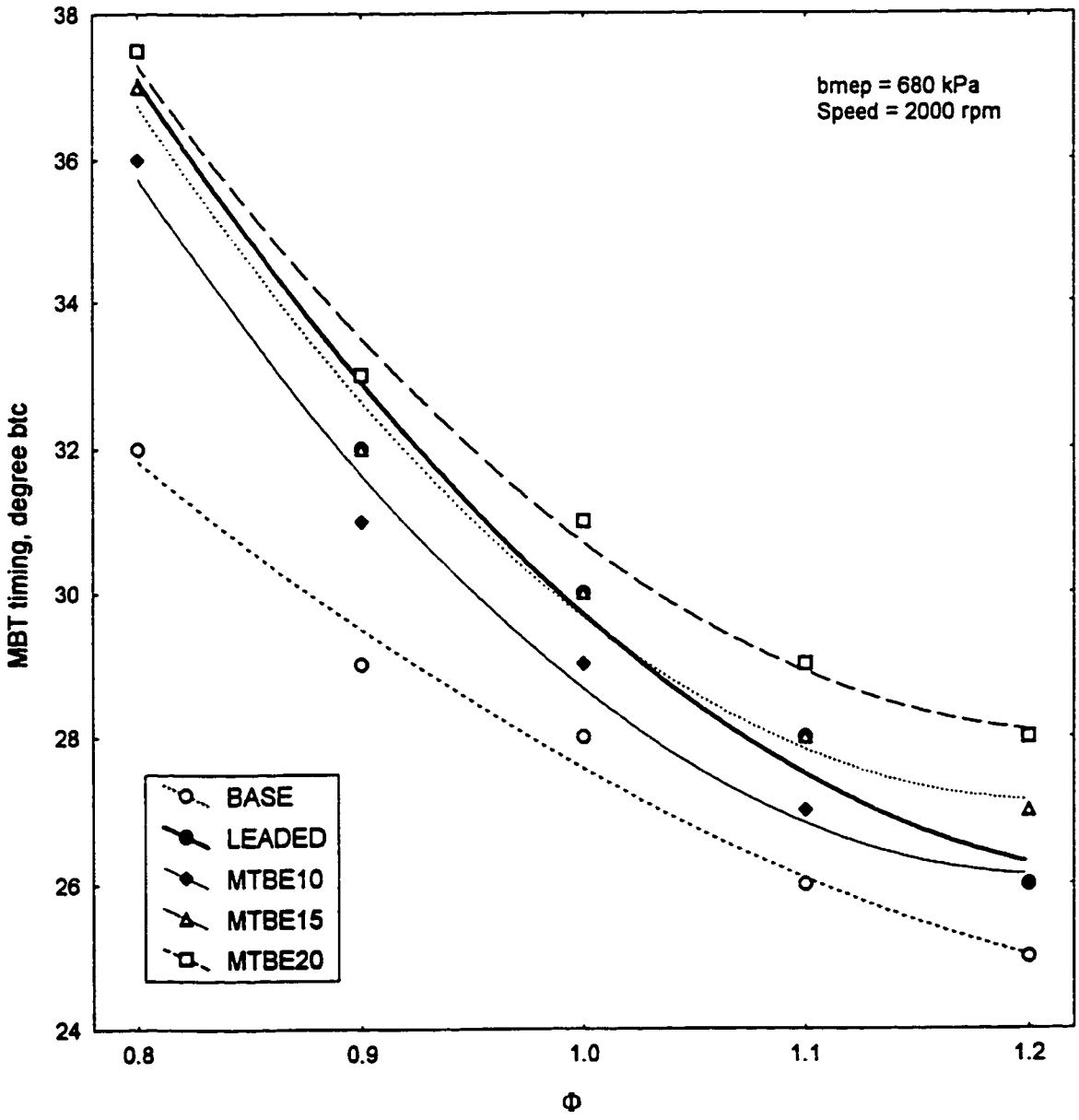

Figure 4-16: Effect of equivalence ratio on MBT timing for the MTBE blends.

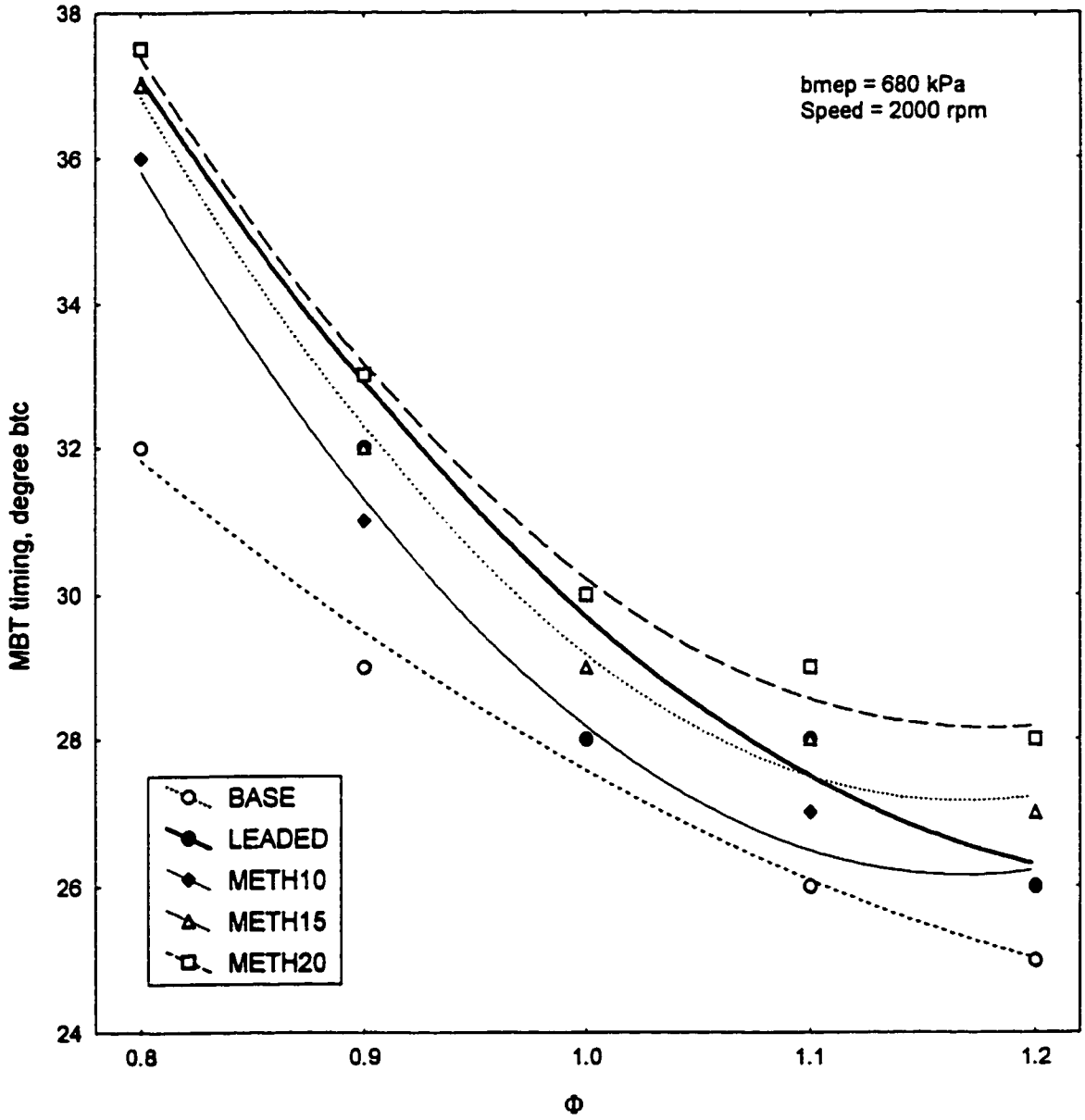

Figure 4-17: Effect of equivalence ratio on MBT timing for the methanol blends.

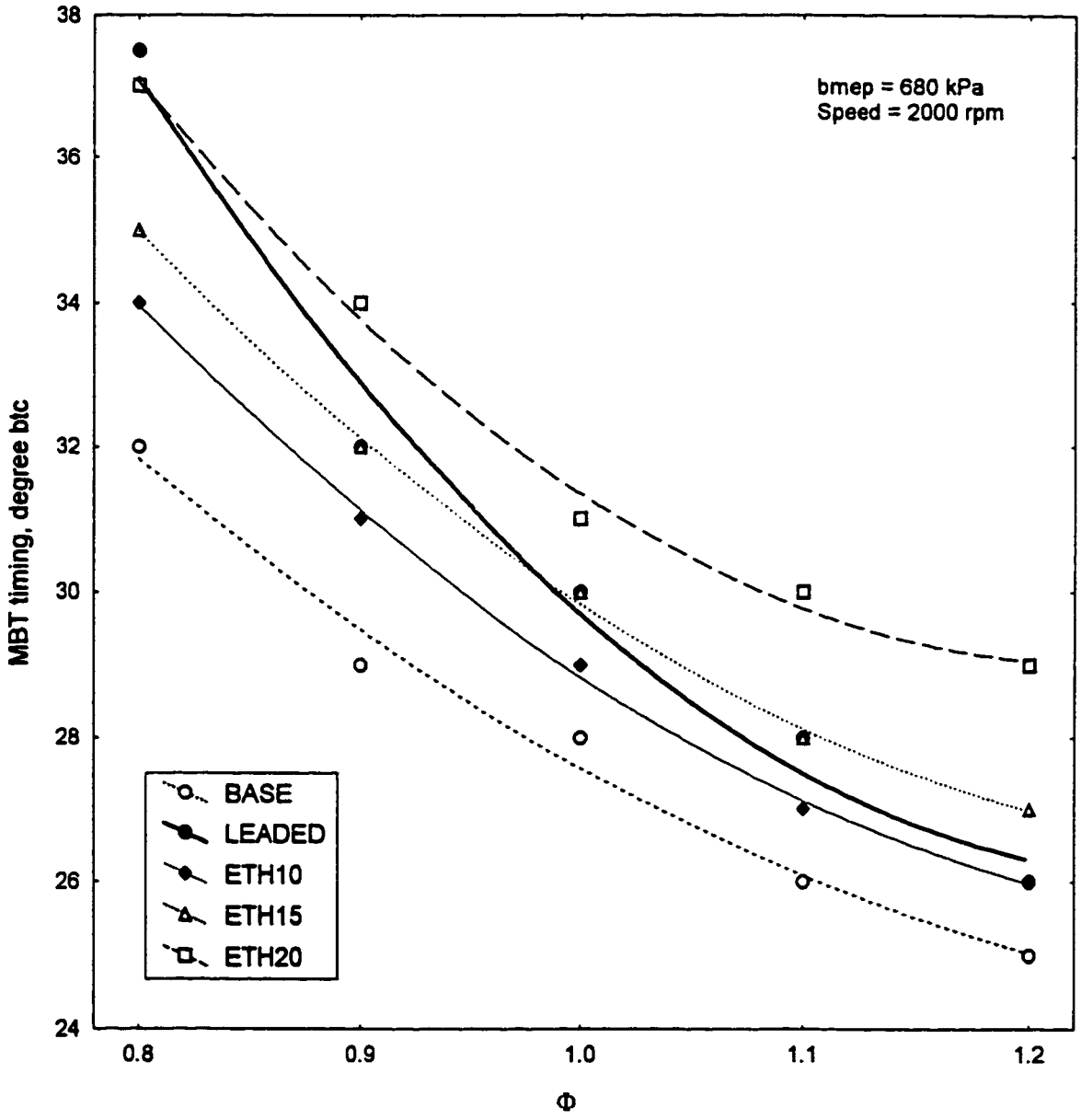

Figure 4-18: Effect of equivalence ratio on MBT timing for the ethanol blends.

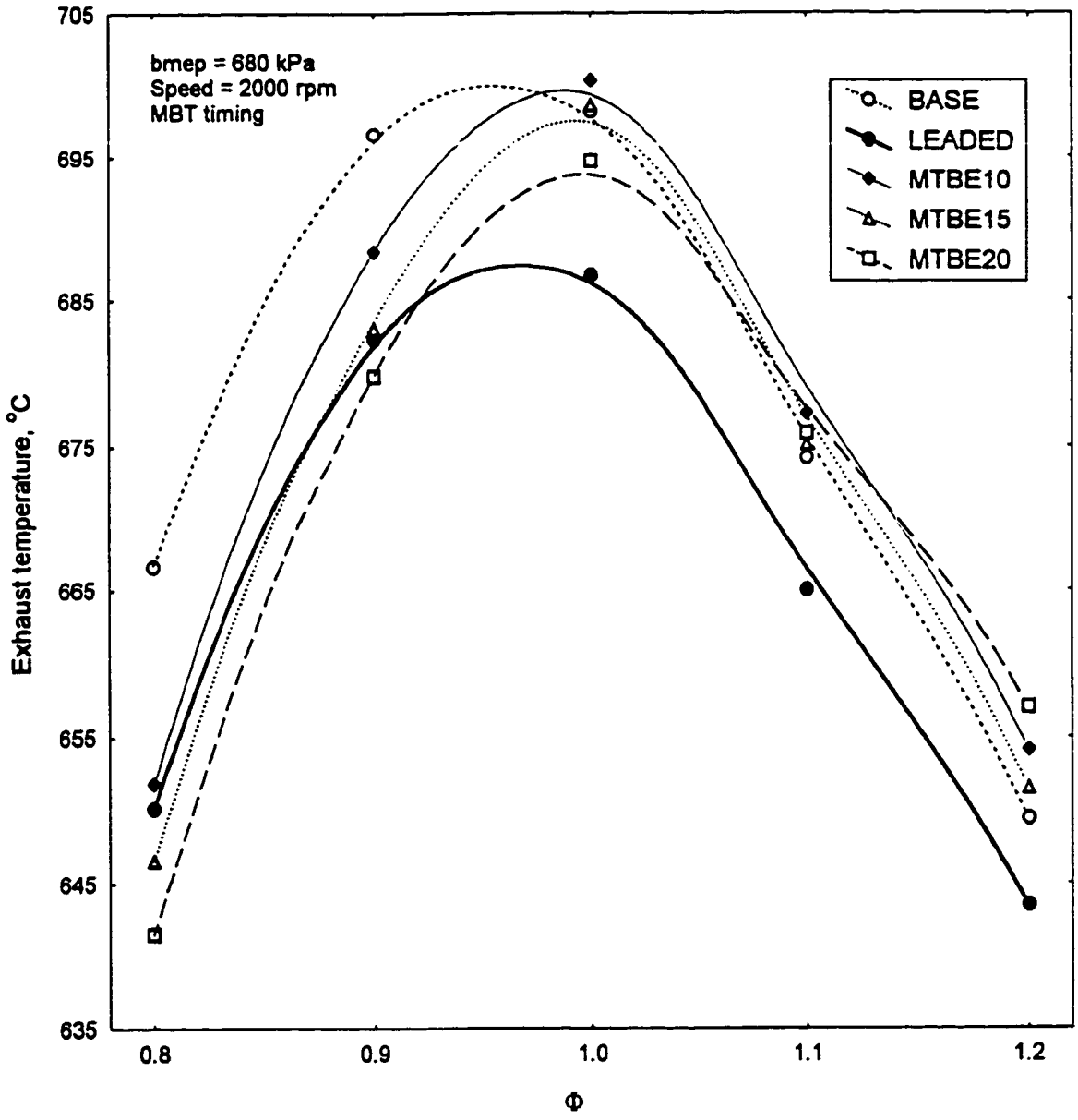

Figure 4-19: Effect of equivalence ratio on exhaust gas temperature for the MTBE blends.

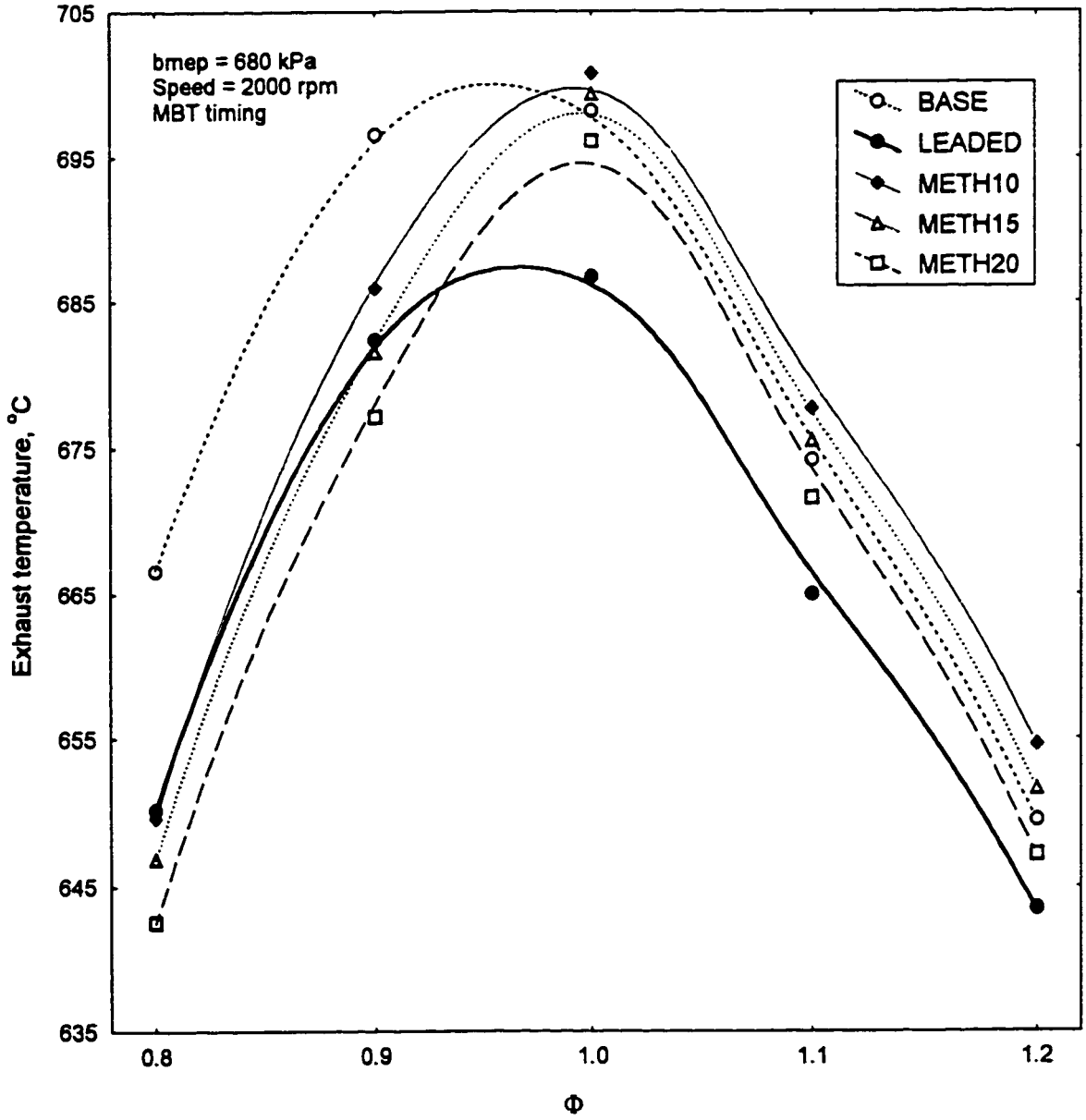

Figure 4-20: Effect of equivalence ratio on exhaust gas temperature for the methanol blends.

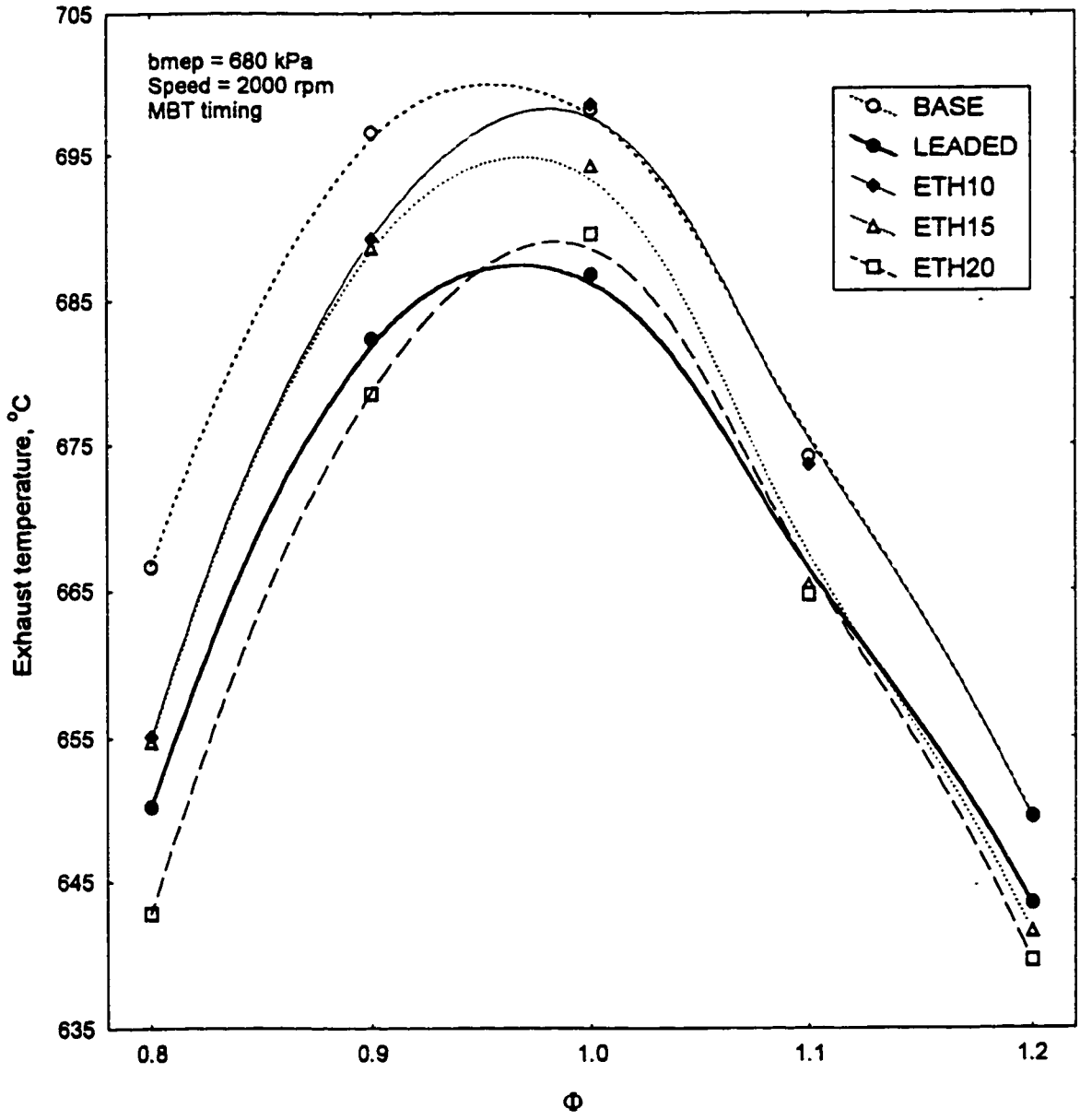

Figure 4-21: Effect of equivalence ratio on exhaust gas temperature for the ethanol blends.

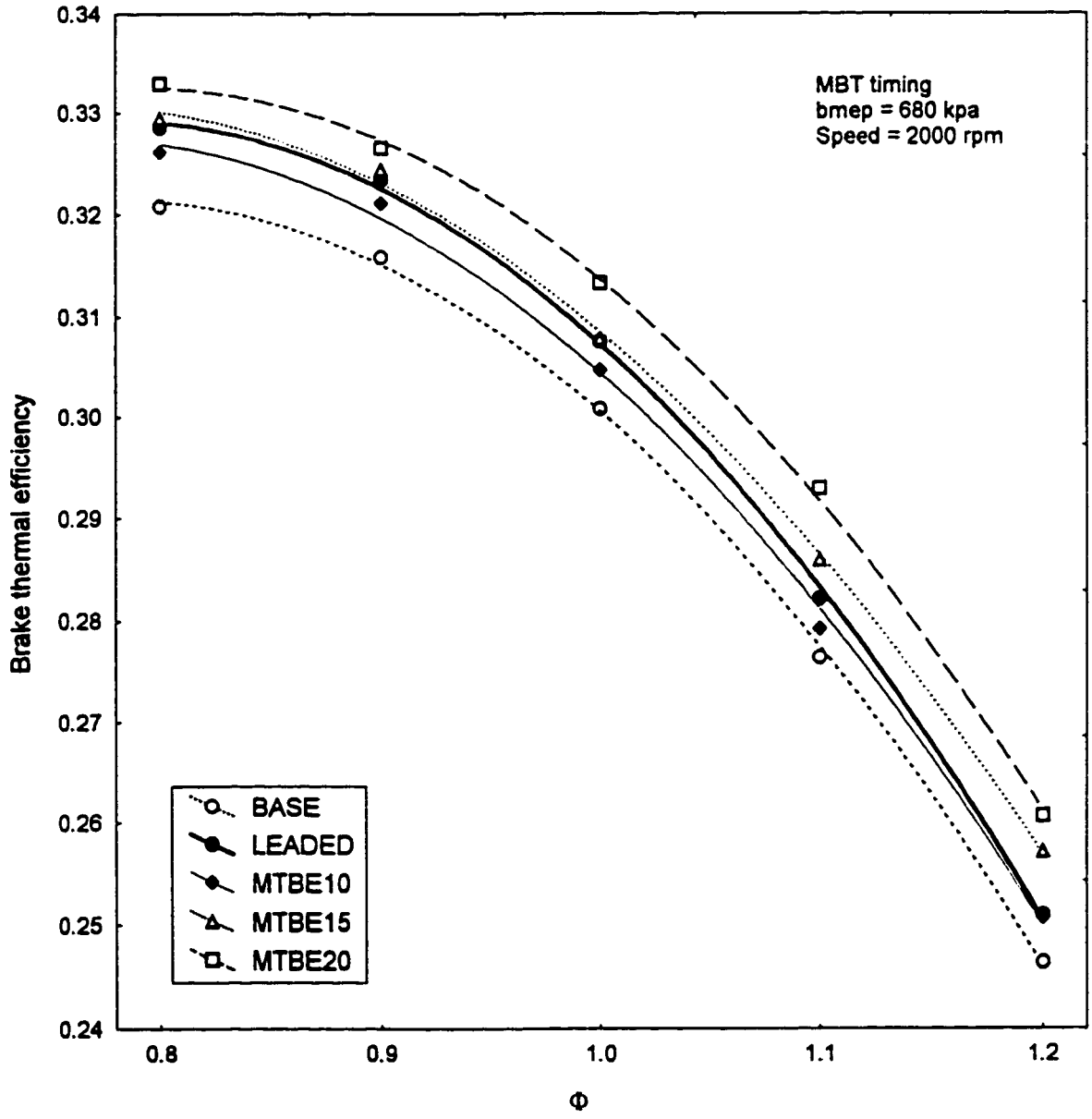

Figure 4-22: Effect of equivalence ratio on brake thermal efficiency for the MTBE blends.

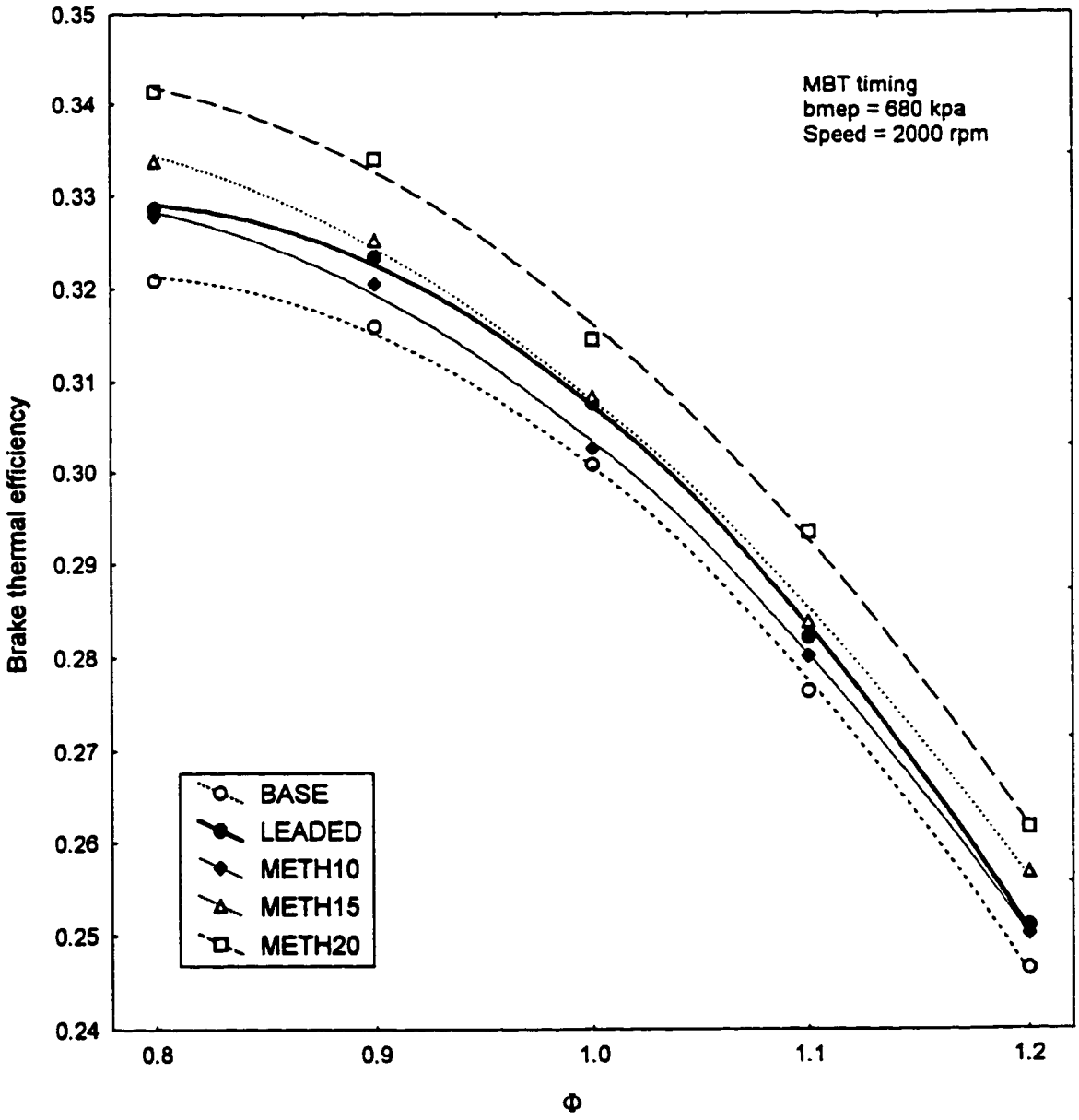

Figure 4-23: Effect of equivalence ratio on brake thermal efficiency for the methanol blends.

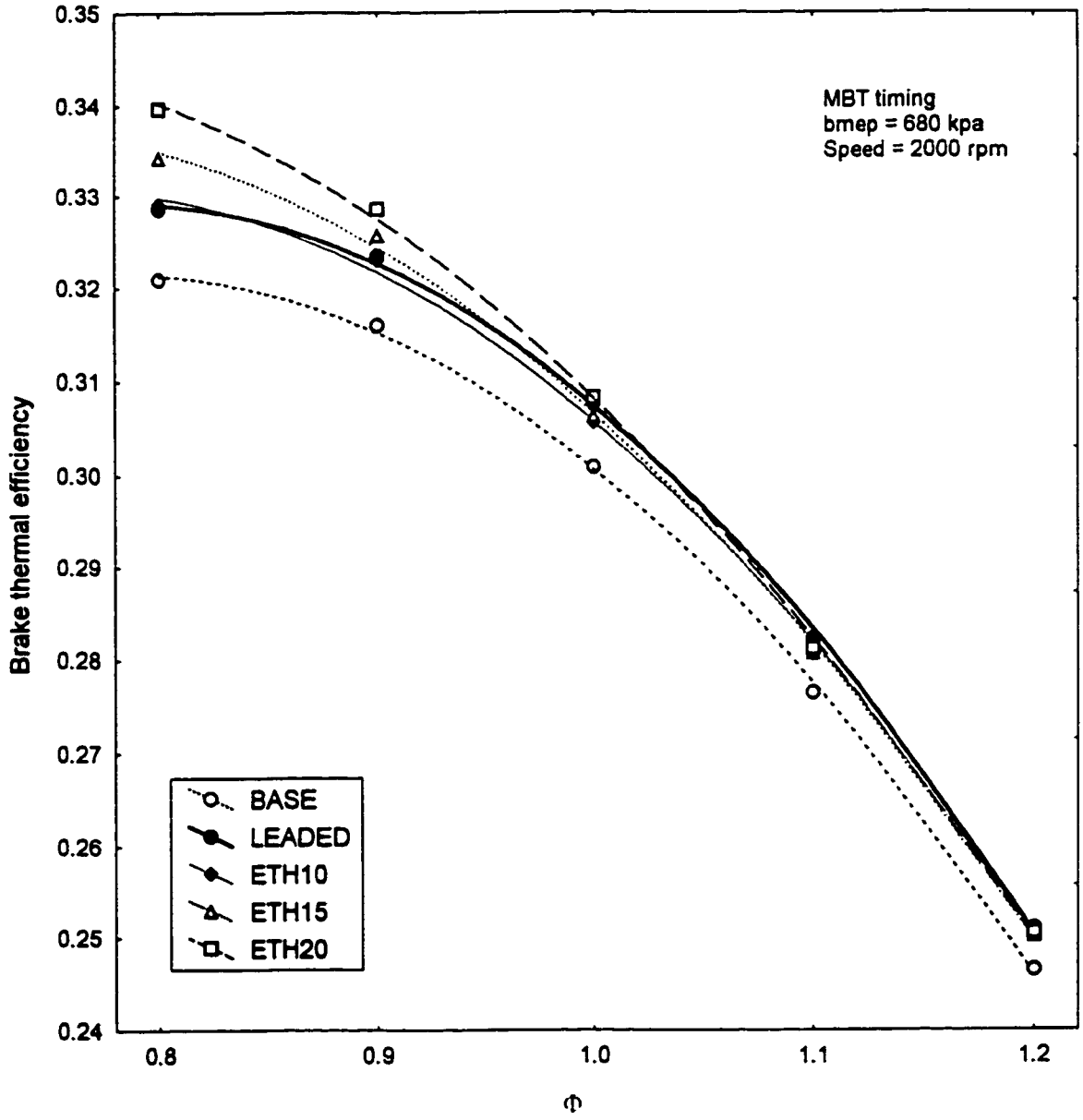

Figure 4-24: Effect of equivalence ratio on brake thermal efficiency for the ethanol blends.

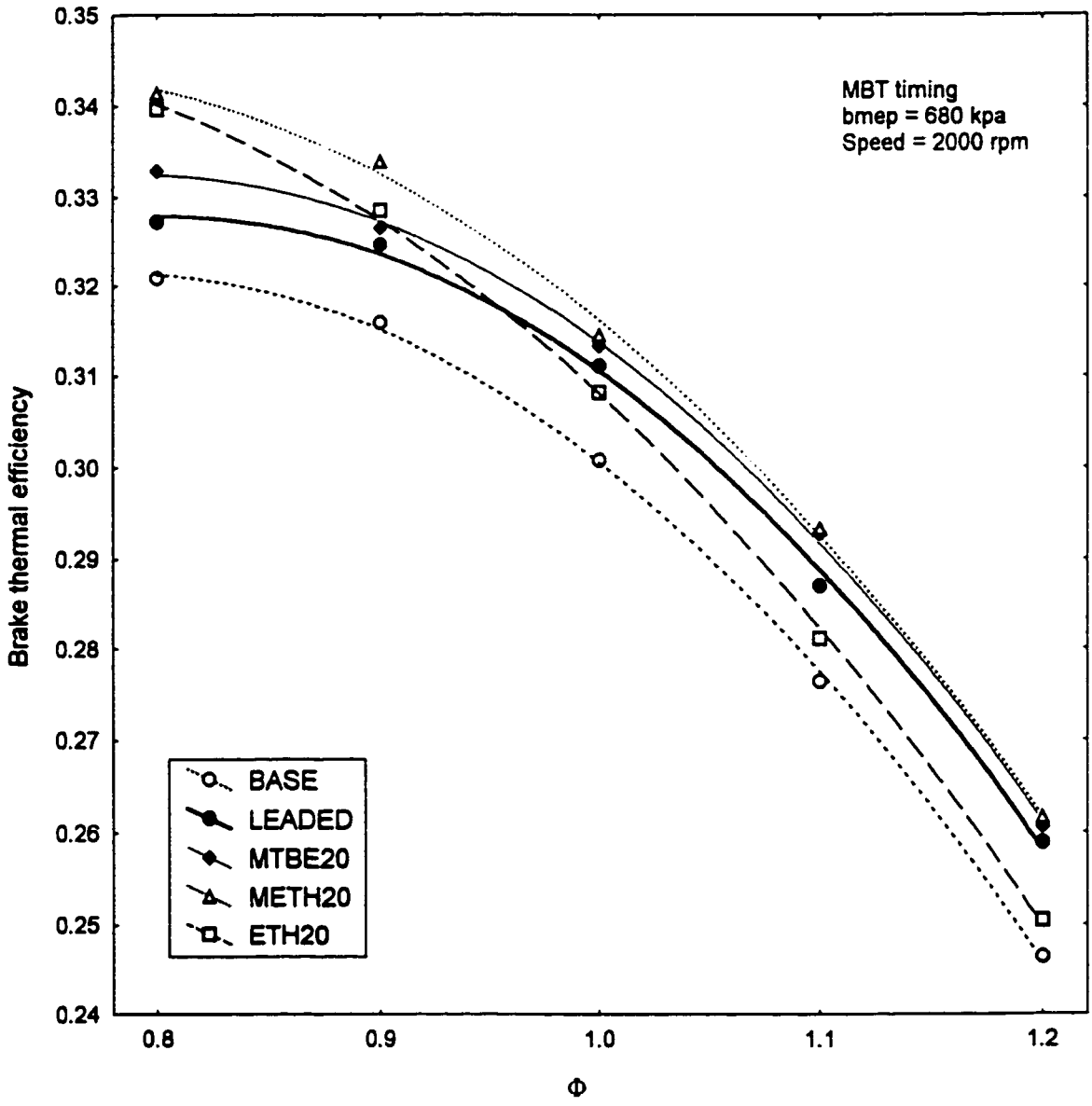

Figure 4-25: Effect of equivalence ratio on brake thermal efficiency for the best performing blends compared to the base and leaded fuels.

Chapter
5

Emissions Results and Discussion

In this chapter, the exhaust emissions from the test engine operating with the test fuels at a variety of engine operating conditions will be presented and discussed. The discussion will emphasize the description of the variation of the emissions with respect to the tested variable and between the different blends. Possible explanations for these variations will be given whenever necessary and available.

5.1 Tested Operating Variables

The exhaust emissions were measured at different conditions that span the practical operating range of normal automotive engines. The effects of most influential operating variables on CO, HC, and NOx emissions were tested. These variables are the equivalence ratio, spark timing, speed, and load. The exhaust emissions were also measured at the idle speed.

The effect of equivalence ratio on exhaust emissions was evaluated at a constant engine speed of 2000 rpm, constant load of 680 kPa, and MBT timing. In these tests, the equivalence ratio ranged from the very lean mixture at Φ=0.8 to the very rich mixture at Φ=1.2. The MBT timing values and exhaust temperatures corresponding to the equivalence ratio tests were presented and discussed in the previous chapter (Figures 4-16 to 4-21).

The effect of spark timing on CO, HC, and NOx emissions was evaluated at constant speed of 2000 rpm, constant load of 680 kPa, and stoichiometric mixtures (Φ=1.0). At these conditions, the spark timing ranged from the very late timing of 10 degrees before the top center (btc) to the early timing of 35 degrees btc.

The effects of the engine speed and load on CO, HC, and NOx emissions for the various test blends were also evaluated. Throughout the speed and load tests, the engine was operating with a stoichiometric mixture and MBT timing. The engine speed spanned a range from 1000 to 3000 rpm, and the tests were conducted at four different loads. The four loads, in terms of brake mean effective pressure (bmep), were: 340, 510, 680, and 850 kPa. The MBT timing values corresponding to the speed and load tests are shown in Figures 5-1 to 5-12. The MBT timing advances as the engine speed increases to compensate for the decreasing time corresponding to the crankshaft degree. In other words, to ensure that the combustion process finishes at the right time in the expansion process so that maximum work transfer to the piston is attained, the combustion must start earlier. For the same fuel, the MBT timing retards toward the top center as the load increases as shown in Figures 5-13, 14, and 15 (for 2000 rpm). This is due to the fact that, as the load increases, more air-fuel mixture is allowed into the engine resulting in higher combustion temperatures and pressures and thus decreasing the combustion time. Because the combustion takes less time, the start of the combustion must be retarded to ensure maximum positive work transfer. Due to their better anti-knock properties, the addition of oxygenates resulted in more advanced MBT timing.

The exhaust emissions of CO, HC, and NOx were measured at the idle speed where the engine operates with practically no applied external load. The engine speed was kept at 1000 rpm and very small load was applied to the engine in order to stabilize its operation. Since different fuels have different optimum spark timing at idle and because there is no clear criterion to determine this optimum timing for each fuel, the exhaust emissions at idle speed were measured at various spark timings. The spark timing is usually used to control the exhaust emissions at idle speed. The results from tests similar to this test can be used to optimize the spark timing so that lower harmful exhaust emissions are achieved at idle speed. The spark timing ranged from 3 to 15 degrees btc and the mixture was kept at stoichiometric throughout the test.

5.2 CO Emissions

In the following sub-sections, the results of the CO emissions for the different blends at the different operating conditions will be presented and discussed. Except in the case of lean mixtures, the results generally indicate a noticeable reduction in CO emissions due to the addition of the oxygeantes over the whole range of spark timing, speed and load. This reduction in CO emissions due to the addition of oxygenates was reported also by many researchers [25, 28, 29, 30, 31, 32, 33, 35, 36].

5.2.1 Effect of Equivalence Ratio on CO Emissions

The results for all the test fuels show a similar trend of CO variation with respect to the equivalence ratio. As the mixture is enriched, the volumetric concentration of CO emissions in the exhaust gases increases sharply and almost linearly. This sharp increase in CO concentration is due to the increasing deficiency of oxygen. As the mixture is leaned out, on the other hand, the CO concentration drops quickly to a relatively very small value (about 0.08 vol.%). Further leaning seems to have no effect on the CO concentration. The excess oxygen in the case of lean mixtures allows more complete oxidation of carbon molecules and thus results in lower CO concentration [2, 4, 5, 11].

Figure 5-16 shows the CO emissions for the MTBE blends as a function of equivalence ratio. For stoichiometric and rich mixtures, the CO concentration decreases noticeably with increasing ratio of MTBE in the blend. For lean mixtures, the addition of MTBE has no effect on CO concentration, which is already very small. The results for the methanol blends, shown in Figure 5-17, indicate a noticeable decrease in CO concentration for rich and stoichiometric mixtures in the case of METH10. Further addition of methanol did not result in further reduction in CO concentration. The results for ethanol blends, shown in Figure 5-18, indicate a slight decrease in CO concentration. The decrease becomes more noticeable as the mixture gets richer and the ethanol ratio in the blend increases. Similar to the MTBE blends, both the methanol and ethanol blends had no effect on CO for lean mixtures.

In general, the addition of oxygenates reduced the CO concentration in the exhaust gas, and the reduction became more significant as the equivalence ratio increased. Figure 5-19 shows the results for the least CO emitting blends compared to the base and leaded fuels. The results for the leaded fuel were similar to those of the base at the whole test range. The CO results for the three oxygenated blends shown in the figure were comparable in the case of stoichiometric and lean mixtures. For rich mixtures, however, the MTBE20 and METH20 blends, which have close results, resulted in more significant reduction than the ETH20 blend.

5.2.2 Effect of Spark Timing on CO Emissions

In general, the results for all the tested fuels show an approximately linear increase in CO concentration as the spark timing is advanced. This is due to the fact that, as the spark timing is advanced towards the optimum timing (MBT timing), the combustion process ends earlier in the expansion stroke allowing more work transfer to the piston and more heat transfer to the cylinder walls resulting in lower exhaust gas temperatures. The decreased exhaust temperature hinders the after-combustion oxidation of CO emissions during the exhaust process and the later stages of expansion. The result is a higher concentration of exhaust emitted CO. Figures 5-20, 21, and 22 show the exhaust gas temperatures as a function of spark timing. The results show a sharp decrease in exhaust temperature as the spark timing is advanced towards the MBT timing [2, 11].

The addition of the three types of oxygenates caused the CO concentration to decrease significantly. The CO concentration continued to decrease as the oxygenate ratio in the blend increases. This can be attributed to the fact that the existence of oxygen as a part of the oxygenate structure may facilitate the oxidation process of the fuel carbon and thus decreasing the concentration of partially oxidized carbon [40]. The most significant decrease in CO emissions is noticed in the case of the ethanol blends. Figures 5-23, 24, and 25 show the results of CO emissions versus spark timing for the MTBE, methanol,

and ethanol blends respectively. Figures 5-26 shows the test results for the least CO emitting blends of the three oxygenates compared to the base and leaded fuels. The results for the leaded fuel are very close to those of the base. Overall, the ETH20 was the least emitting blend with a maximum reduction of about 17% compared to the base fuel. The least emitting methanol blend, which is METH20, resulted in a maximum reduction of about 13% and the least emitting MTBE blend, which is MTBE20, in 8%.

5.2.3 Effects of Speed and Load on CO Emissions

The results show that the CO concentration increased, in most cases, linearly as the engine speed increased. This is despite the increasing exhaust gas temperature, as indicated in Figures 5-27, 28, and 29 (for bmep = 680 kPa), which normally promotes further oxidation of partially burned fuel carbon. In this case, the effect of increasing exhaust temperature seems to be more than offset by the decreasing residence time in the exhaust manifold due to the increasing gas flow rate [2, 11]. The results show a consistent jump of CO concentration at engine speed of 1500 rpm. No clear explanation is available for this behavior. However, it has been mentioned earlier in chapter 4 that the noticed fluctuation in the engine brake torque at lower speeds was linked to the abnormal vibration of the engine test cell. This abnormal engine vibration at certain speeds may play a role also in the formation of exhaust emissions. The engine vibration may affect the intensity of gas turbulence inside the combustion chamber and therefore affect the combustion quality and products.

The CO results for the MTBE blends are shown in Figures 5-30 to 5-33. The reduction of CO emissions due to the addition of MTBE is sustained over the entire range of speed and load. The variation of CO concentration versus the load at a constant speed of 2000 rpm is shown in Figure 5-34. Similar trend is noticed for the methanol blends, Figures 5-35 to 5-39, and the ethanol blends, Figures 5-40 to 5-44. The results for the least CO emitting blends at the different loads are shown in Figures 5-45 to 5-48. At lower loads (340-510 kPa), the MTBE20 blend was the least CO emitting in general with a

maximum reduction with respect to the base fuel of about 24% at bmep = 340 kPa and 18% at bmep = 510 kPa. At higher loads (680-850 kPa), the least CO emitting blend was the ETH20 with a maximum reduction of about 12.6% at bmep = 680 kPa and 11.8% at bmep = 850 kPa. The leaded fuel generally resulted in higher CO emissions than the oxygenated blends.

5.2.4 CO Emissions at Idle Speed

The results show that the CO concentration increases sharply as the spark timing is advanced. This increase can be linked to the decrease in exhaust gas temperature shown in Figure 5-49. With lower exhaust temperature, less carbon will be fully oxidized in the after-combustion reactions resulting in higher CO concentration [2].

Figures 5-50, 51, and 52 show the results for the MTBE, methanol, and ethanol blends respectively. The addition of the three oxygenates resulted in significant reduction in CO concentration particularly as the spark timing becomes closer to the top center. The addition of 10 vol.% of MTBE or methanol resulted in a very significant reduction of CO at spark timing close to top center. However, the effect on CO diminishes as the spark timing is advanced. Further addition of MTBE or methanol sharply decreased the CO concentration at advanced spark timing but had less significant effect at spark timing close to the top center. For the ethanol blends, increasing the ethanol ratio caused an increasingly more significant reduction in CO emissions.

The results for the least CO emitting blends are shown in Figure 5-53. A maximum reduction of about 35% in CO with respect to the base fuel is achieved by MTBE20. The results for METH20 came close to those of MTBE20, while ETH20 resulted in less significant reduction in CO particularly at advanced spark timing. The CO results for the leaded fuel are lower than those of the base fuel at late spark timing (close to the top center), but the results for the two fuels become closer to each other as the spark timing is advanced.

5.3 HC Emissions

The results for the HC emissions, presented in the following sub-sections, generally indicate that the oxygenated blends decreased HC emissions with respect to the base and leaded fuels except at very high loads and at very advanced spark timing. The reduction in HC emissions due to the addition of oxygenates was reported also by many researchers [22, 23, 28, 29, 31, 32, 33, 36].

5.3.1 Effect of Equivalence Ratio on HC Emissions

The effect of equivalence ratio on the HC volumetric concentration, measured in ppm C (part per million as single-carbon-atom compound), is qualitatively similar for all the tested fuels. The HC concentration is minimum at about Φ=0.9 and rises with increasing or decreasing the equivalence ratio. As the equivalence ratio is increased from 0.9, the HC concentration increases sharply and almost linearly. As the equivalence ratio is decreased from 0.9, the HC concentration increases slowly.

The high HC concentration in the case of rich mixtures is a result of insufficient oxygen at the time of combustion. The HC concentration decreases considerably as the mixture is leaned out. However, after a certain limit of leaning, the combustion quality deteriorates and the HC concentration rises again [2, 4, 5, 11].

Figure 5-54 shows the results for the MTBE blends. For Φ=0.9 and more, a slight decrease is noticed in HC concentration as the MTBE ratio in the blend is increased. For Φ=0.8, however, the addition of MTBE resulted in a slight increase in HC concentration. The results for the methanol blends, shown in Figure 5-55, show no significant differences with respect to the base fuel. The results for the ethanol blends show, in general, a slight increase in HC concentration with respect to the base fuel, as shown in Figure 5-56.

The results for the least HC emitting blends, in this test, compared to the base and leaded fuels are shown in Figure 5-57. The leaded fuel resulted in a noticeably higher HC concentration than the base fuel. This may be a result of the lower exhaust

temperature noticed in the case of the leaded fuel (see Figures 4-19, 20, and 21). Lower exhaust temperature results in less effective after-combustion oxidation of unburned hydrocarbons, which leads to higher exhaust HC concentration [2]. In general, all the blends performed better than the leaded fuel except at very rich mixtures (Φ=0.8). Among the oxygenated blends, the least HC emitting blend was the MTBE20 with noticeably lower HC concentration than the base fuel in the equivalence ratio range of 0.9 to 1.2.

5.3.2 Effect of Spark Timing on HC Emissions

The results for all the test fuels show an increase in HC concentration with advancing spark timing. Similar to the case of CO emissions, the increase in HC emissions observed with advancing spark timing is due to the decreased after-combustion temperatures. The reduced temperature in exhaust stroke and later stages of expansion diminishes the chance of unburned hydrocarbons to continue the reaction with available oxygen [2, 11].

Figures 5-58, 59, and 60 show the results for the MTBE, methanol, and ethanol blends respectively. In general, the addition of MTBE and methanol reduced the HC concentration. This reduction is most significant at very late spark timing (10 degrees btc) and becomes less significant as the spark timing is advanced. The reduction due to the addition of MTBE is much significant than that of methanol. The addition of up to 15 vol.% of ethanol caused a slight reduction in HC concentration at late spark timing but increased it at advanced timing. Figure 5-61 shows the test results for the least HC emitting blends of the three oxygenates compared to the base and leaded fuels. Overall, the MTBE20 blend is by far the least HC emitting blend with a maximum reduction of about 20% with respect to the base fuel at very late spark timing. However, the differences between the blends are less important at very advanced timing. The leaded fuel caused an increase in HC concentration with respect to the base fuel.

5.3.3 Effects of Speed and Load on HC Emissions

The results show a fluctuation in HC concentration with respect to the engine speed at lower loads (340 and 510 kPa). As the load is increased, however, the intensity of this fluctuation weakens and eventually vanishes at the highest tested load (850 kPa).

This fluctuation is most severe at speed range between 1500 and 2000 rpm. The only explanation found for this fluctuation is the abnormal mode of engine vibration experienced at this range of speed. A measurement of the engine vibration, in terms of average noise level, as a function of speed at full load was given earlier in chapter 4. The vibration measurement, shown in Figure 4-10, indicates a fluctuation in the intensity of vibration as the engine speed increases. The fluctuation of vibration in the speed range between 1500 and 3000 rpm can be correlated with the fluctuation of HC concentration. This correlation can be explained by the possible effect of vibration on the intensity of gas turbulence inside the combustion chamber. More vibration is expected to promote the gas turbulence and thus improve the mixing quality between the fuel and the air. Improving the mixing quality results in better combustion and less unburned HC.

The results suggest that the influence of the engine vibration on the HC emissions weakens as the load is increased. This can be attributed to the increasing influence of other factors such as the combustion pressure and temperature and the exhaust temperature.

Except for the lowest tested load (340 kPa) and the speed range from 1000 to 1500 rpm, the HC concentration generally decreases as the engine speed increases. This decrease can be attributed, at least partly, to the corresponding sharp increase in exhaust temperature. It has been stated previously, however, that the effect of exhaust temperature in this case is offset by the decreasing residence time in the exhaust manifold due to the increasing flow rate.

Figures 5-62 to 5-65 show the results for the MTBE blends. The addition of MTBE resulted in lower HC concentration except at the highest load (850 kPa) where the

addition of 10 vol.% caused an increase with respect to the base fuel, particularly at lower speeds, and further addition of MTBE had no effect.

The effect of methanol on HC concentration is similar to that of MTBE, as shown in Figures 5-66 to 5-69, except at the highest load. At this load, the addition of 10 vol.% methanol increased the HC concentration at mid-range speed and slightly decreased it at very low and very high speeds. Furthermore, increasing the methanol ratio had no effect at mid-range and high speeds but significantly decreased the HC concentration at low speeds.

The results for the ethanol blends are shown in Figures 5-70 to 5-73. These results show that the addition of ethanol reduced the HC concentration for the first two loads (340 and 510 kPa). For higher loads, the addition of ethanol increased the HC concentration except for some cases where a decrease is noticed with the ETH20 blend.

Figures 5-74 to 5-77 compare the results for the least HC emitting blends with those of the base and leaded fuels. For the first three loads (up to 680 kPa), the least emitting blend is the MTBE20. The difference between the blends becomes more evident as the load is increased. For the highest load, the lowest HC concentration resulted from the METH20 at lower speeds and from the MTBE20 at higher speeds. The results for the leaded fuel are comparable to those of the base with slight increase or decrease in some cases.

The HC concentrations versus load at a constant speed of 2000 rpm for all the blends are shown in Figures 5-78 to 5-81. These results indicate that the reduction of HC emissions due to the addition of oxygenates is maximum at lower loads. As the load is increased (at constant speed), the HC emissions from the base decrease while increasing from the oxygenated blends. At higher loads, the HC emissions from the oxygenated blends are either comparable or higher than that of the base fuel. The best performing blend with respect to the HC emissions is clearly the MTBE20.

5.3.4 HC Emissions at Idle Speed

In general, the exhaust HC concentration at idle speed increases sharply and linearly as the spark timing is advanced. This increase can be linked to the decrease in exhaust gas temperature observed with advancing spark timing (Figure 5-49). The lower exhaust gas temperature results in less after-combustion oxidation of the fuel carbon and hydrogen.

Figures 5-82, 83, and 84 show respectively the results for the MTBE, methanol, and ethanol blends. The addition of oxygenates reduced the exhaust HC concentration significantly. In the case of the MTBE blends, the reduction in HC emissions was almost proportional to the MTBE ratio in the blend. In the case of the methanol blends, the most significant reduction is noticed with 10 vol.% methanol (METH10). Further increase of methanol ratio caused a further, but less significant, reduction in HC concentration. The significance of the reduction in the case of the methanol blends is most at very late (close to the top center) and very advanced spark timing. In contrast to the methanol, the addition of 10 vol.% ethanol (ETH10) caused only a slight reduction of HC emissions. Further addition of ethanol resulted in noticeable reduction with respect to the base fuel particularly at advanced spark timing.

The results for the least HC emitting blends compared to the base and leaded fuels are shown in Figure 5-85. The METH20 had the lowest concentration of HC emissions among the blends with a maximum reduction of about 31% with respect to the base fuel. The leaded fuel resulted in higher HC emissions than the base fuel except at very late spark timing (close to the top center) where a lower concentration is noticed.

5.4 NOx Emissions

The results presented in the following sub-sections indicate that the addition of oxygenates generally resulted in higher NOx emissions except in the case of the 10 vol.% ethanol and in the case of lean mixtures. The addition of 10 vol.% of MTBE or methanol caused the NOx to increase when compared to the base fuel. But while increasing MTBE

resulted in more NOx emissions, increasing the methanol ratio in the blend caused the NOx to decrease. On the contrary, the addition of 10 vol.% ethanol resulted in less NOx, but increasing the ethanol caused the NOx to increase or decrease depending on the conditions. These complicated effects may explain some of the conflict in the published results regarding the effect of oxygenates on NOx emissions. Some researchers reported an increase of NOx emissions as a result of adding oxygenates [29, 32, 36] and others reported a decrease [22, 26, 28] or no effect [28, 29, 31]. It must be emphasized, however, that these published results have been taken at different test conditions and with different procedures and equipment as indicated in the literature review.

5.4.1 Effect of Equivalence Ratio on NOx Emissions

The results from this test show a similar trend of variation of NOx emission concentration with respect to the equivalence ratio for all the tested fuels. The maximum NOx concentration (measured in volumetric ppm) is encountered at about Φ=0.9. Increasing or decreasing the equivalence ratio from 0.9 causes the NOx concentration to decrease. The formation of NOx emissions is mainly controlled by the combustion temperature and the availability of oxygen. As these two parameters increase, more NOx is formed. The combined effect of these parameters is highest at Φ=0.9, therefore the maximum NOx concentration is attained there [2, 4, 5,11].

Figure 5-86 shows the results for the MTBE blends. For stoichiometric and rich mixtures, the MTBE blends resulted in higher NOx concentration than the base fuel. For lean mixtures ($\Phi \leq 0.9$), however, the addition of MTBE resulted in a significant decrease in NOx concentration. In general, increasing the MTBE ratio in the blend resulted in further slight increase in NOx concentration except for very lean mixtures where, in contrast, a significant decrease is noticed.

The results for the methanol blends, shown in Figure 5-87, generally indicate an increase in NOx concentration with respect to the base fuel except for very lean mixture (Φ=0.8) where a decrease is noticed. Close results were obtained for the three methanol

blends which may indicate that the maximum effect of methanol on NOx emissions is attained with small ratios, and further increase of methanol has no significant effect. Another more acceptable explanation, which is supported by other emissions results discussed later, is that the effect of methanol is reversed as its ratio is increased. In other words, the NOx concentration increases with small ratio of methanol but starts to decrease as this ratio increases.

Figure 5-88 shows the results for the ethanol blends. The NOx concentration decreased with the addition of 10 vol.% ethanol (ETH10), but further addition of ethanol caused the NOx to rise again except in the case of very lean mixture (Φ=0.8). This trend is supported by other emissions results discussed later in this chapter. With very lean mixtures, the NOx concentration continued to decrease significantly as the ethanol ratio increased.

The results for the least NOx emitting blends compared to the base and leaded fuels are shown in Figure 5-89. For $\Phi \geq 0.9$, the least NOx emitting blend was the ETH10 with a decrease of up to about 6% with respect to the base fuel. The results for the leaded fuel were very close to those of the base.

5.4.2 Effect of Spark Timing on NOx Emissions

Generally, the results show a sharp and approximately linear increase in NOx concentration as the spark timing is advanced. This is due to the increasing combustion pressures and temperatures inside the combustion chambers of the engine with advancing spark timing (see Figure 2-3) [2, 5, 11].

Figure 5-90 shows the results for the MTBE blends. The addition of 10 vol.% of MTBE (MTBE10) had no effect on NOx concentration at very late spark timing but resulted in a noticeable increase in NOx with respect to the base fuel at advanced timing (>30 degrees btc). Increasing MTBE ratio up to 20 vol.% resulted in only a slight increase in NOx concentration.

The results for the methanol blends are shown in Figure 5-91. The three methanol blends resulted in higher NOx concentration than the base fuel. The results indicate a significant increase in NOx concentration particularly at advanced timing in the case of the METH10 blend. Increasing the methanol ratio to 15 vol.% (METH15) had no effect on NOx with respect to METH10 except at very late timing (10 degrees btc) where a slight decrease is noticed. The METH20 blend noticeably decreased the NOx concentration with respect to METH10 except at advanced timing where no change is noticed. These results indicate that the effect of methanol on NOx emissions changes as the methanol ratio increases. In other words, the NOx concentration increases with addition of small ratios of methanol but decreases as this ratio increases. This reversal of effect has been discussed earlier in section 5.4.1.

Figure 5-92 shows the results for the ethanol blend. Here, the trend is opposite to that of the methanol blends. Addition of 10 vol.% ethanol (ETH10) resulted in a significant reduction in NOx concentration with respect to the base fuel. Further increase in ethanol ratio increased the NOx concentration except at very late spark timing where no effect is noticed.

The results for the least NOx emitting blends compared to the base and leaded fuels are shown in Figure 5-93. While the MTBE and methanol blends resulted in higher NOx emissions with respect to the base, the ETH10 resulted in a reduction of up to 11%.

5.4.3 Effects of Speed and Load on NOx Emissions

The results show generally a significant increase in NOx emissions with increasing speed. This increase can be attributed to the increasing combustion temperature due to the increasing rate of heating cycles and the decreasing significance of heat transfer. Another factor that affects the peak combustion temperature, and thus the formation of NOx emissions, is the decreasing of residual gases that are left in the combustion chamber from previous cycles as the speed is increased [2, 5, 11].

A fluctuation in the NOx concentration is noticed at the lowest tested load (340 kPa). Similar to the of HC emissions, the fluctuation here may have been influenced by the fluctuation in the engine vibration noticed with increasing speed. However, the nature of this influence in this case is not fully understood. Generally speaking, the engine vibration is expected to influence the combustion quality which in turn influences the combustion temperature. This reasoning is supported by the results of exhaust temperature presented earlier in this section 5.2.3 (Figures 5-27, 28, and 29). In these results, a slight fluctuation in the exhaust temperature is noticed particularly at speed range between 1000 and 2000 rpm. This later fluctuation is compatible with the fluctuation in NOx emissions at lower loads. As the load is increased however, the fluctuations in the NOx emissions vanish.

The results for the MTBE blends are shown in Figures 5-94 to 5-97. These results show that the addition of MTBE caused an increase in the NOx concentration. Higher MTBE ratio resulted in higher NOx emissions. In general, the NOx increase due to the MTBE addition becomes more significant as the load is increased. This is evident in Figure 5-98 that shows the variation of NOx emissions with respect to the load at a constant speed of 2000 rpm.

Figures 5-99 to 5-102 show the results for the methanol blends. A significant increase in NOx emissions is noticed with the addition of 10 vol.% of methanol. Increasing the methanol ratio to 15 vol.% caused additional, but less significant, increase in the NOx concentration. Further increase of methanol to 20 vol.% (METH20) caused a decrease in NOx emissions with respect to the 15 vol.% methanol blend. This reversal of effect has been reported earlier in sections 5-4-1 and 5-4-2. The increase in NOx concentration is most significant at mid-rang speeds and higher loads. The variation of NOx concentration with respect to the load at a constant speed of 2000 rpm is shown in Figure 5-103.

The results for the ethanol blends are shown in Figures 5-104 to 5-107. The addition of 10 vol.% ethanol resulted in lower NOx concentration with respect to the base fuel at all speeds and loads. However, increasing the ethanol ratio generally increased the NOx concentration except at the high speed range of the low load test (340 kPa). This effect, which is opposite to that of the methanol, has been discussed earlier in sections 5-4-1 and 5-4-2. The variation of NOx emissions with respect to the load at a constant speed of 2000 rpm is shown in Figure 5-108.

The results for the least NOx emitting blends compared to the base and leaded fuels at the different loads are shown in Figures 5-109 to 5-112. These results show that the least NOx emitting blend is the ETH10. This blend caused a reduction in the NOx emissions with respect to the base fuel at all test speeds and loads. The other blends, shown in these figures, resulted generally in higher NOx emissions with respect to the base fuel. The results for the leaded fuel showed no clear trend with respect to the base fuel but in most cases resulted in higher NOx emissions.

5.4.4 NOx Emissions at Idle Speed

The results, shown in Figures 5-113, 114, and 115, indicate that the concentration of NOx emissions at idle speed is very small (in the order of 100 ppm) and insignificant. This is due to the low combustion pressures and temperatures characterizing the no-load operation. The variation of NOx concentration at idle speed with spark timing is also insignificant (in the order of 20 ppm). Furthermore, the variation of NOx concentration due to the addition of the oxygenates is in the order of 10 ppm which is very small.

In the light of these results, it can be concluded that the addition of oxygenates has no significant effect on NOx emissions at idle speed, which are already insignificant.

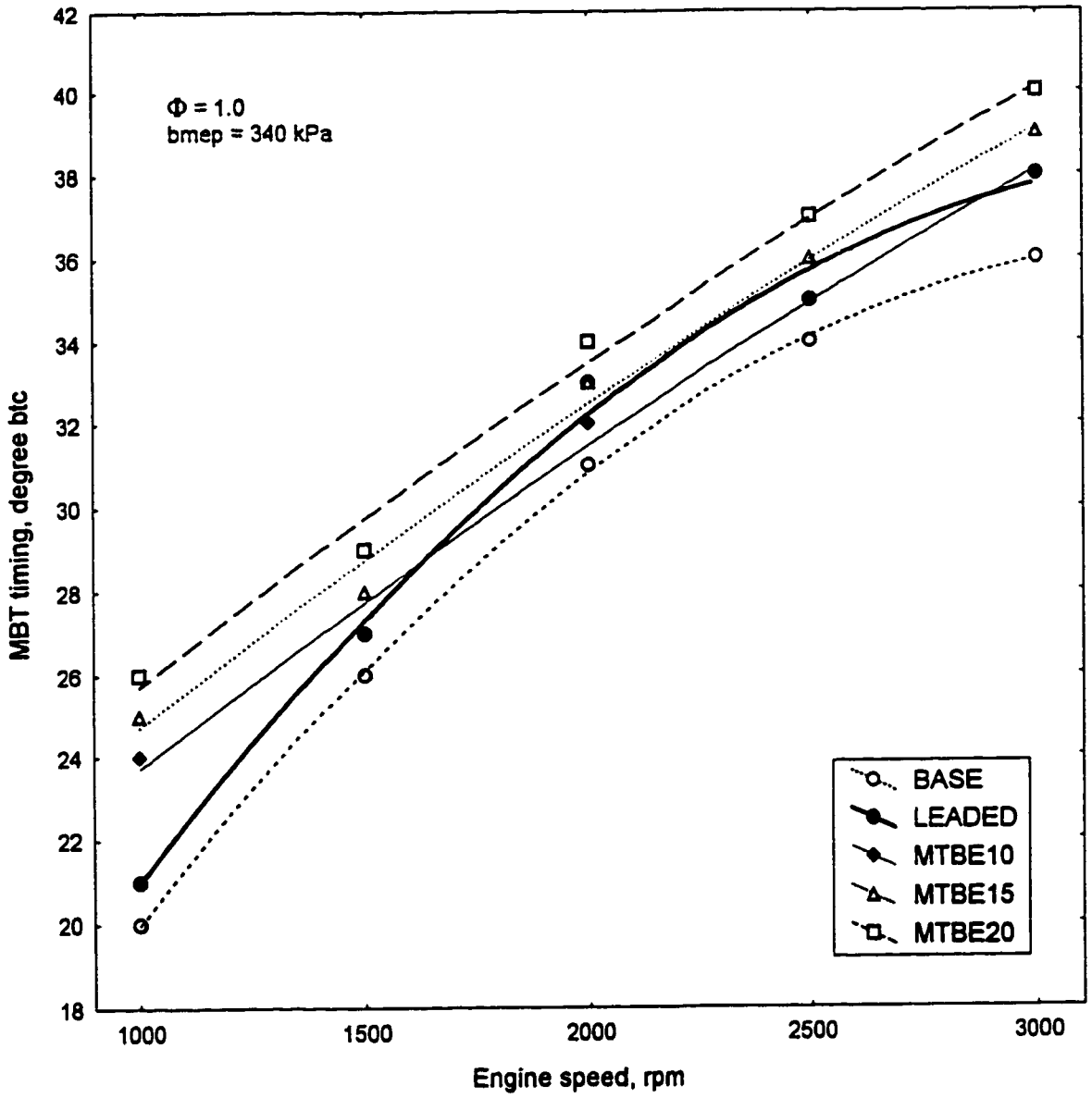

Figure 5-1: MBT timing vs. speed at a constant load of 80 Nm (bmep = 340 kPa) for the MTBE blends.

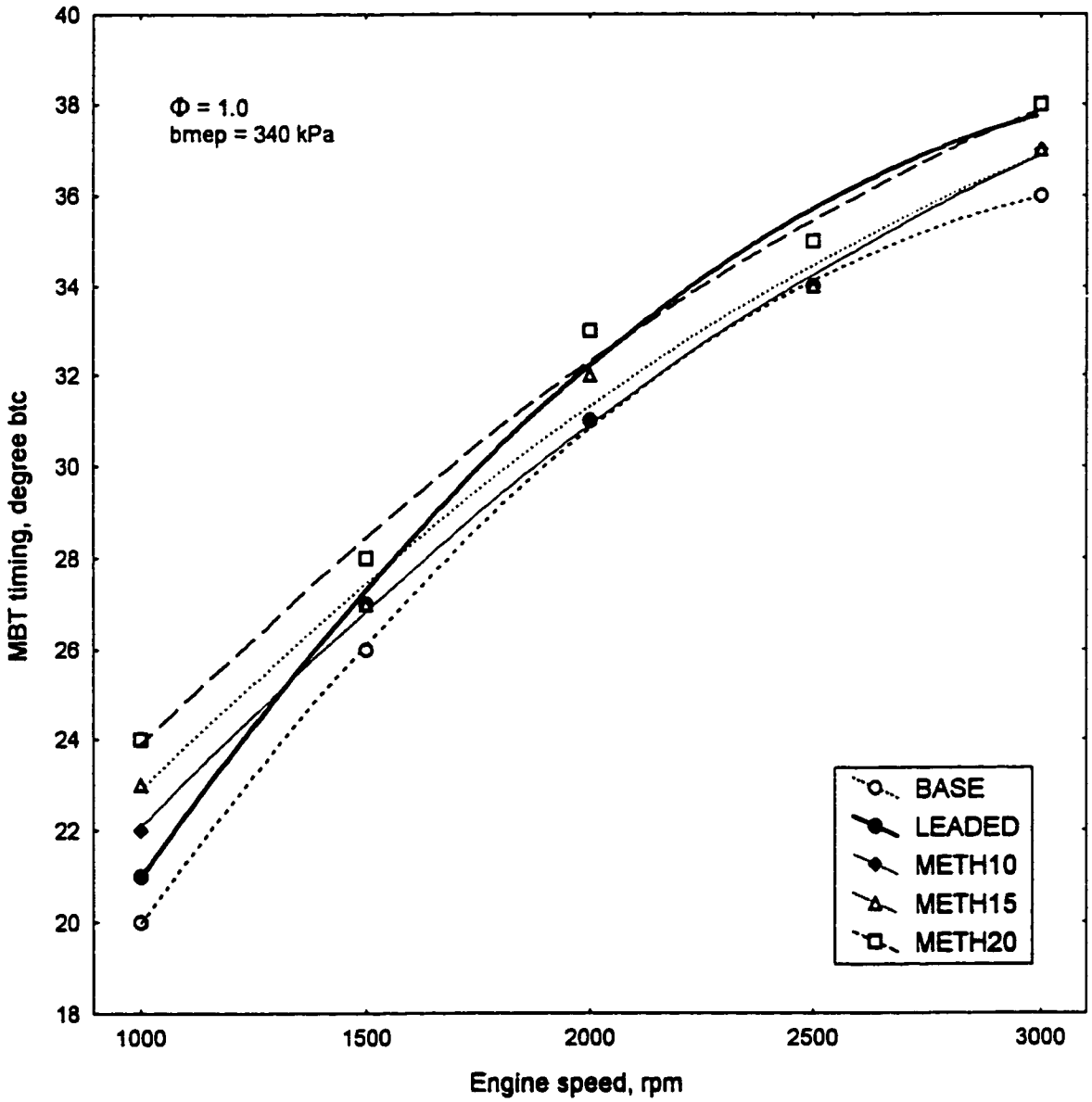

Figure 5-2: MBT timing vs. speed at a constant load of 80 Nm (bmep = 340 kPa) for the methanol blends.

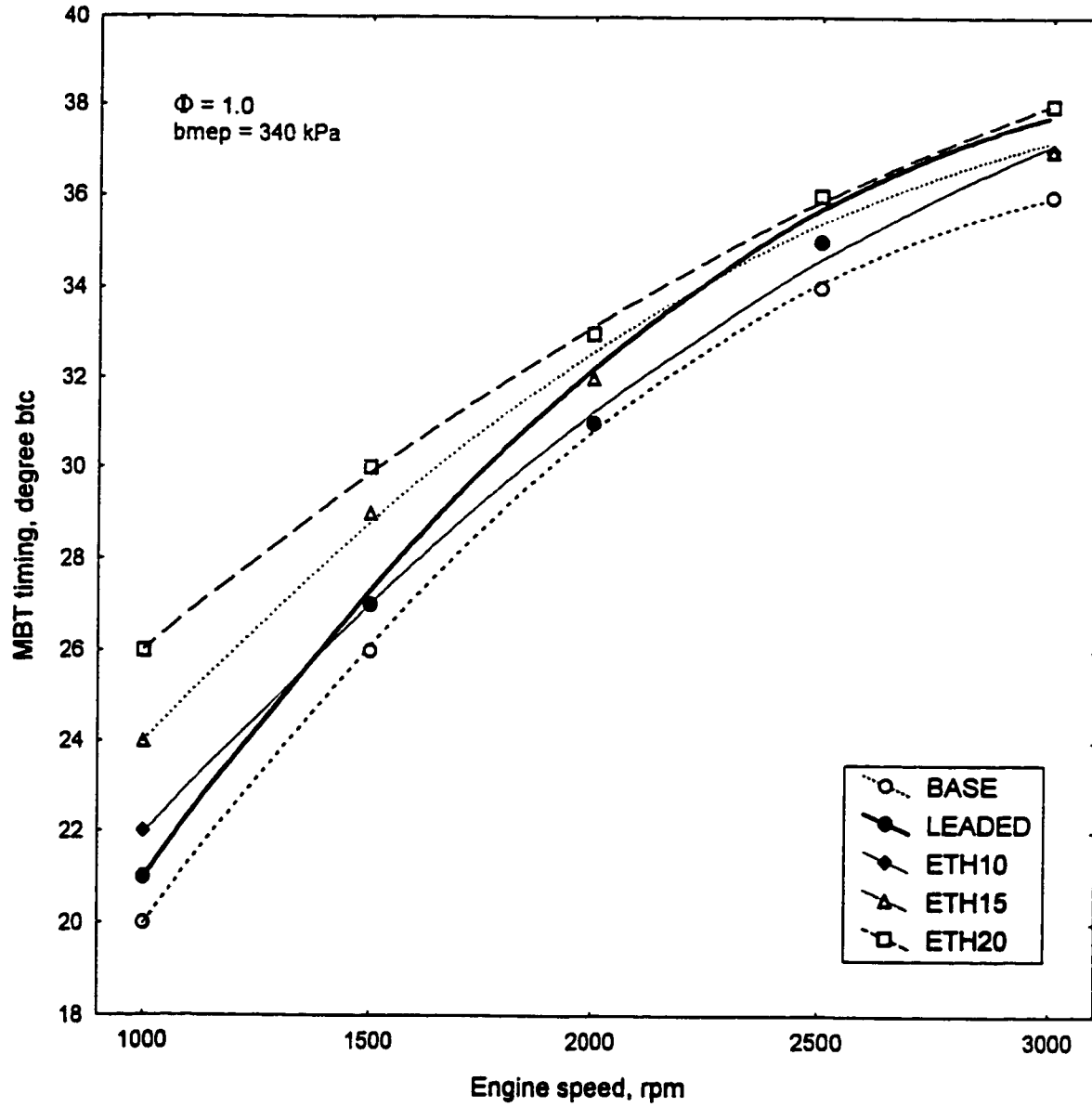

Figure 5-3: MBT timing vs. speed at a constant load of 80 Nm (bmep = 340 kPa) for the ethanol blends.

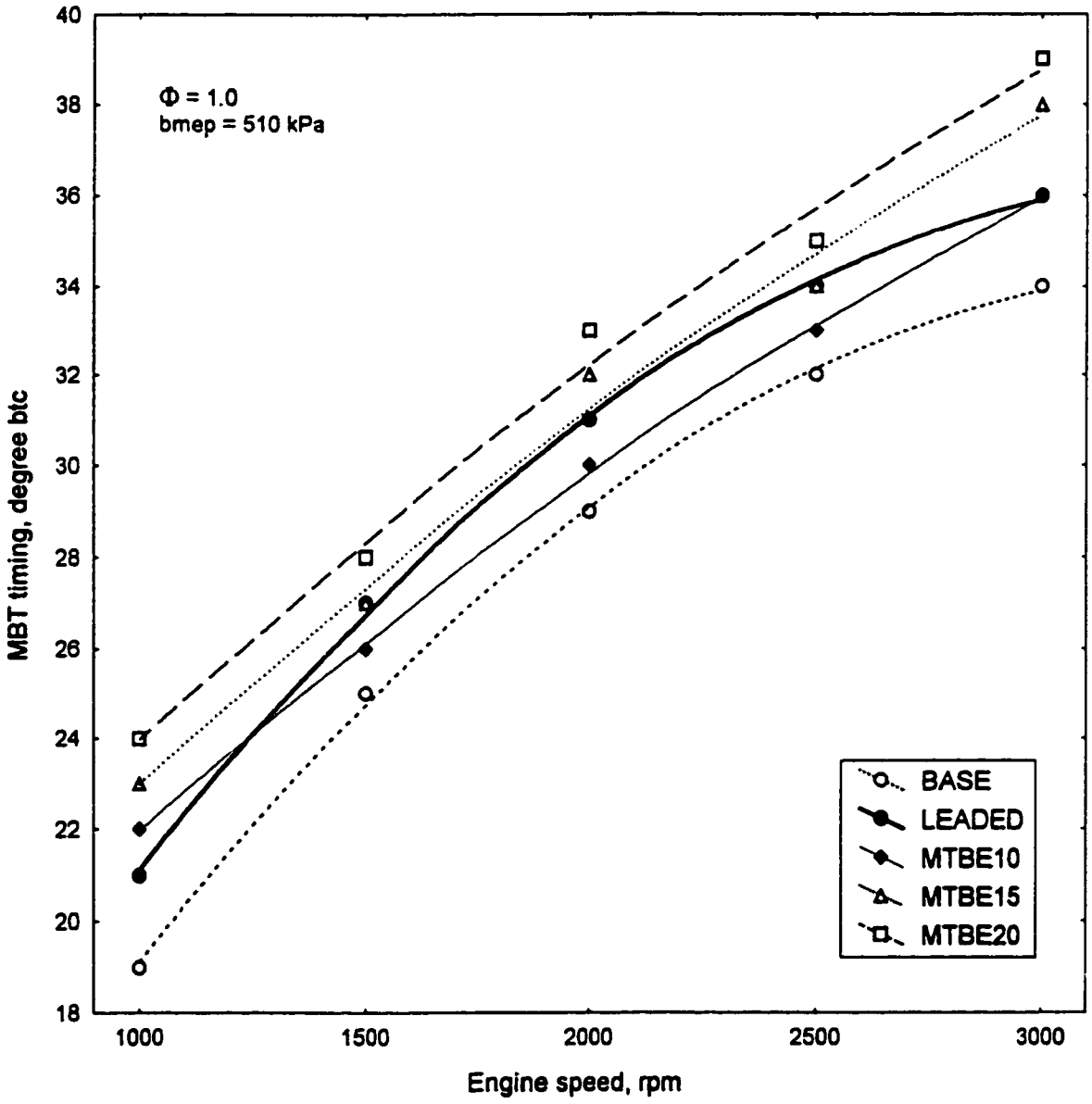

Figure 5-4: MBT timing vs. speed at a constant load of 120 Nm (bmep = 510 kPa) for the MTBE blends.

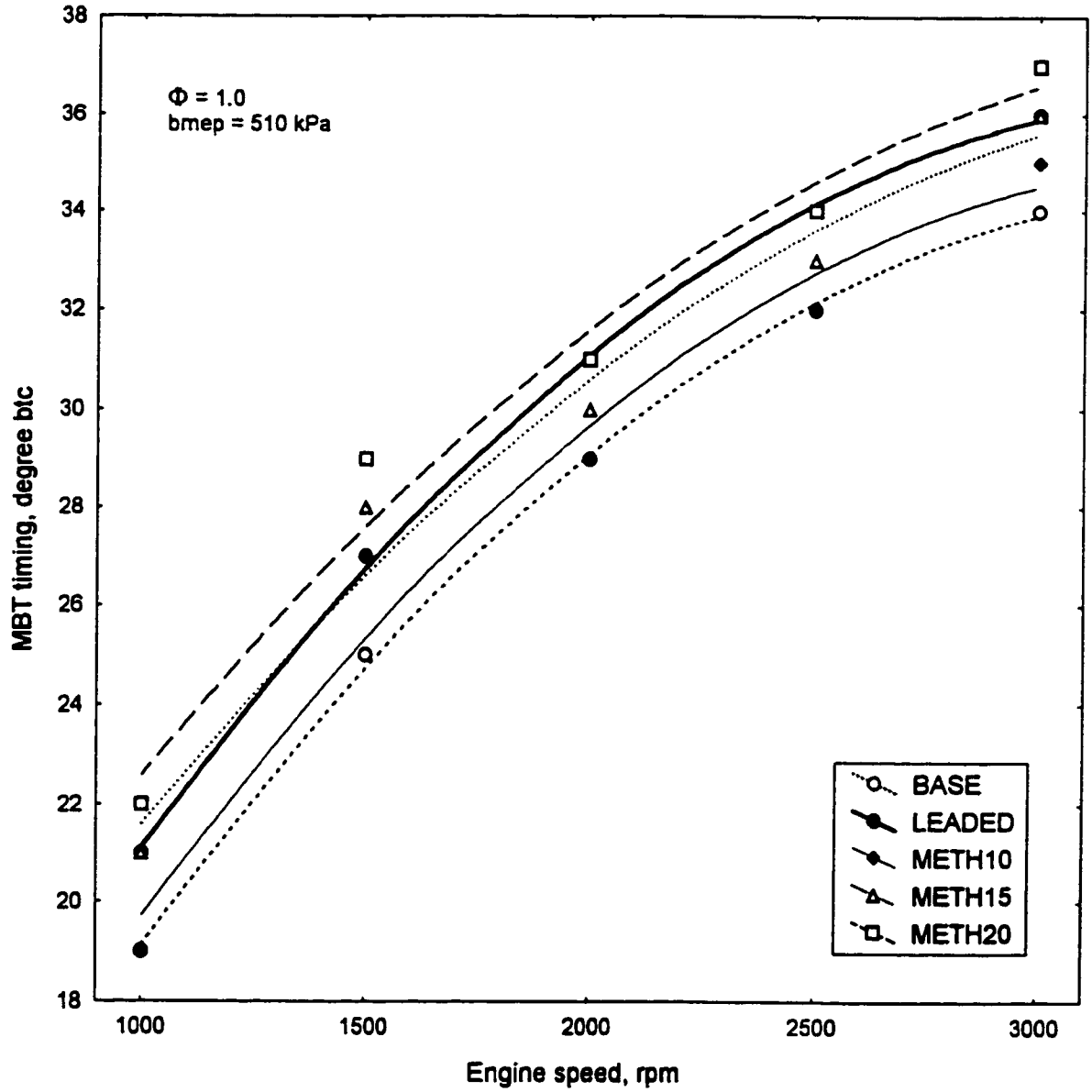

Figure 5-5: MBT timing vs. speed at a constant load of 120 Nm (bmep = 510 kPa) for the methanol blends.

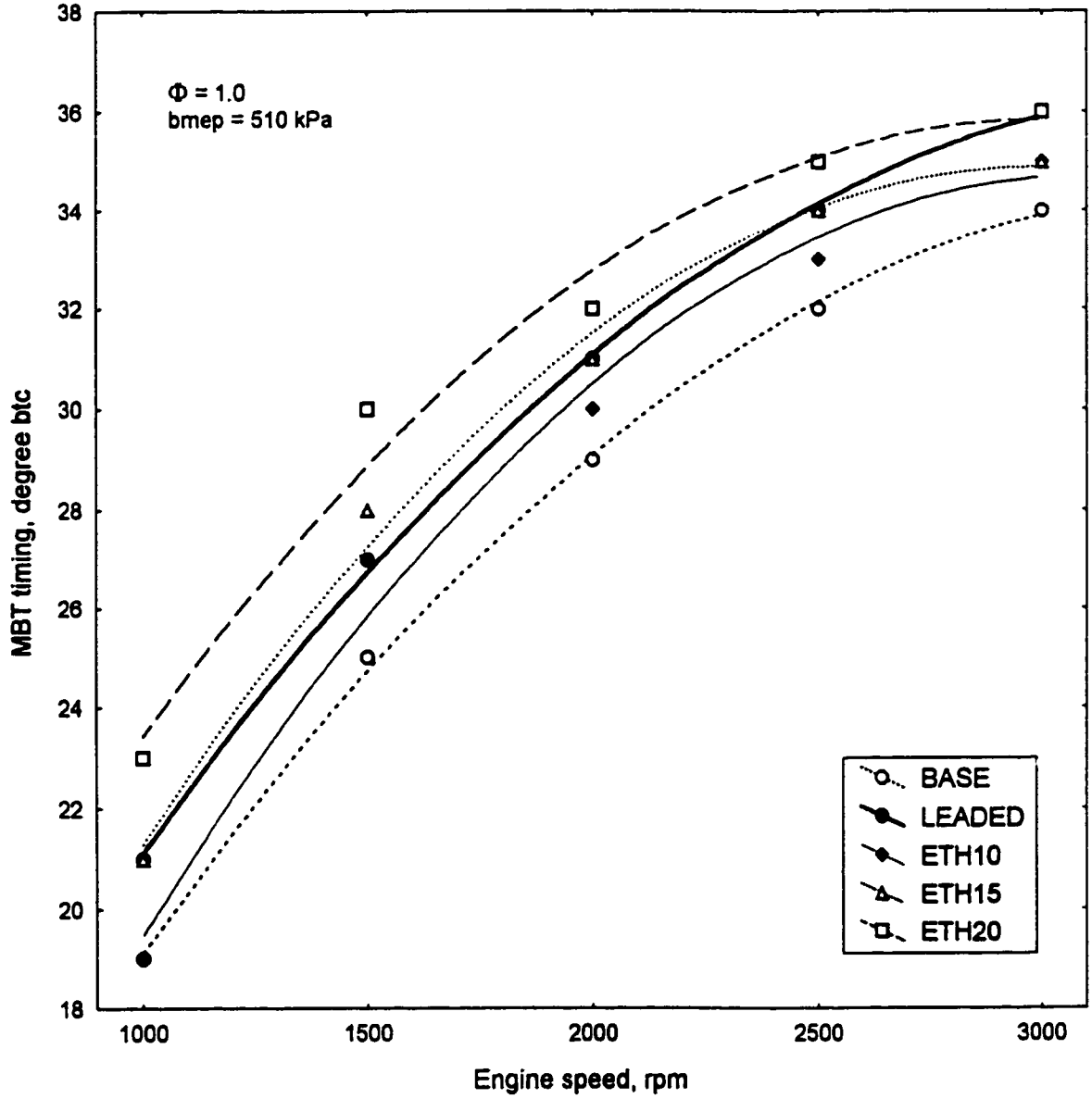

Figure 5-6: MBT timing vs. speed at a constant load of 120 Nm (bmep = 510 kPa) for the ethanol blends.

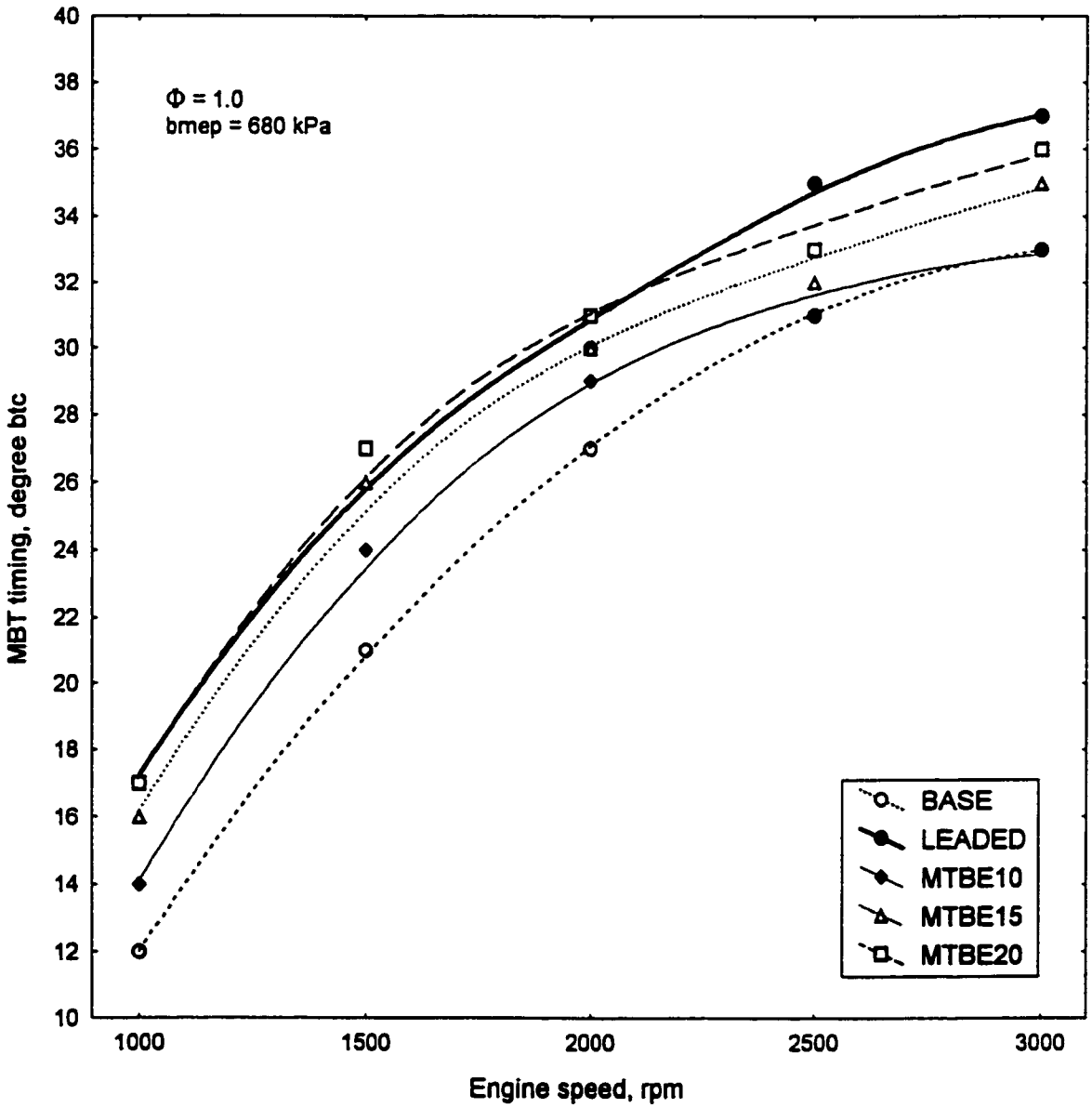

Figure 5-7: MBT timing vs. speed at a constant load of 160 Nm (bmep = 680 kPa) for the MTBE blends.

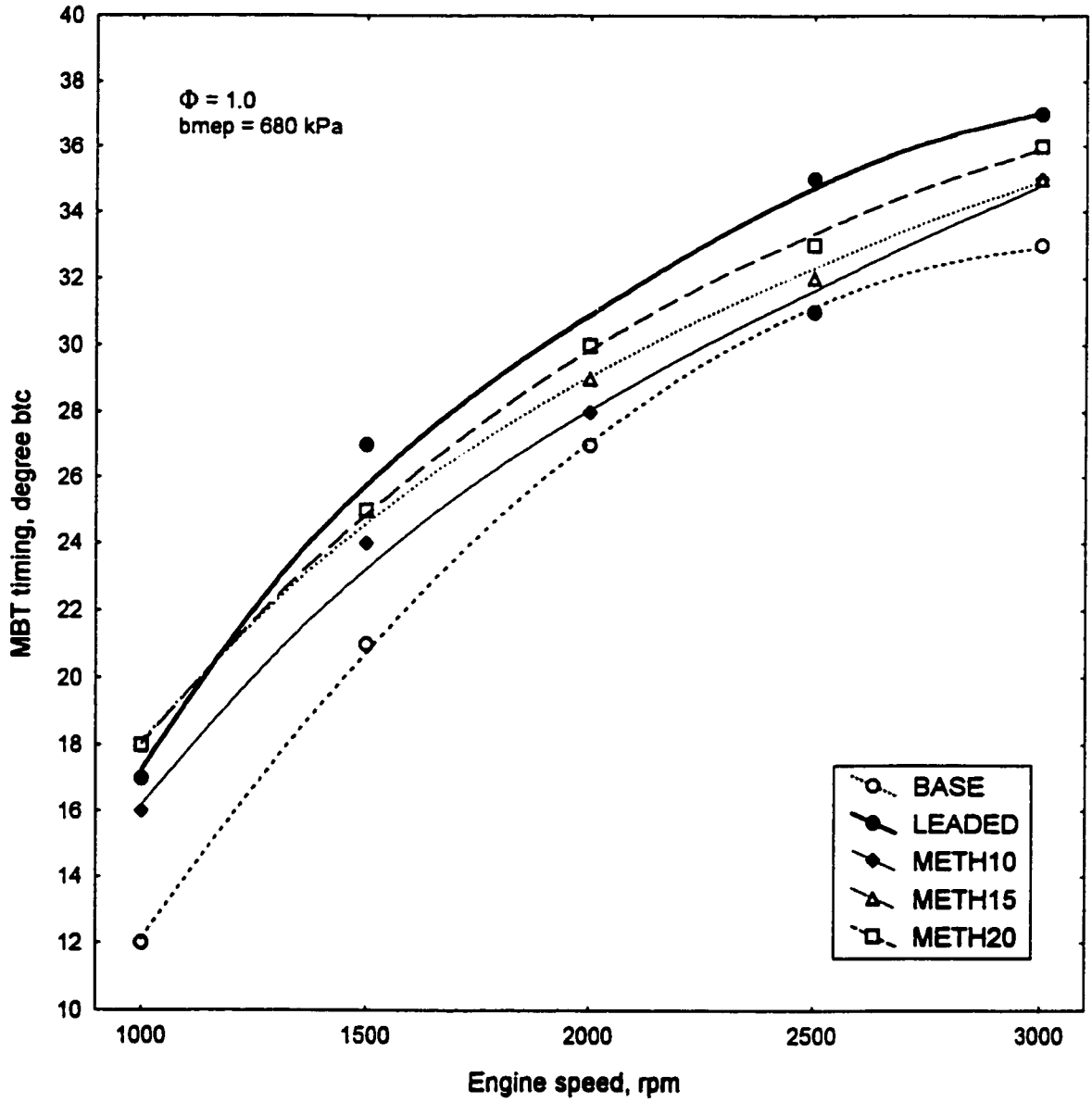

Figure 5-8: MBT timing vs. speed at a constant load of 160 Nm (bmep = 680 kPa) for the methanol blends.

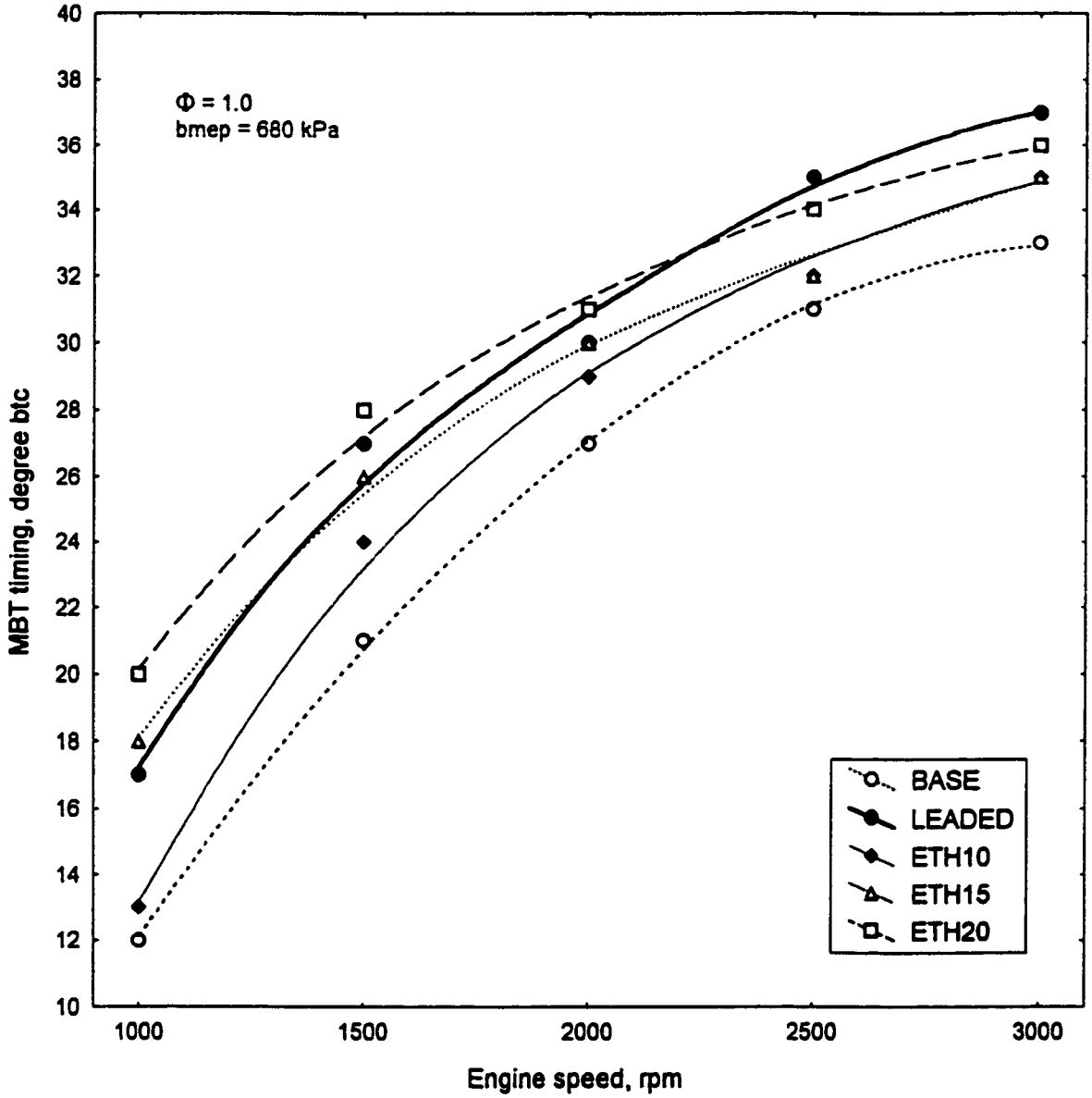

Figure 5-9: MBT timing vs. speed at a constant load of 160 Nm (bmep = 680 kPa) for the ethanol blends.

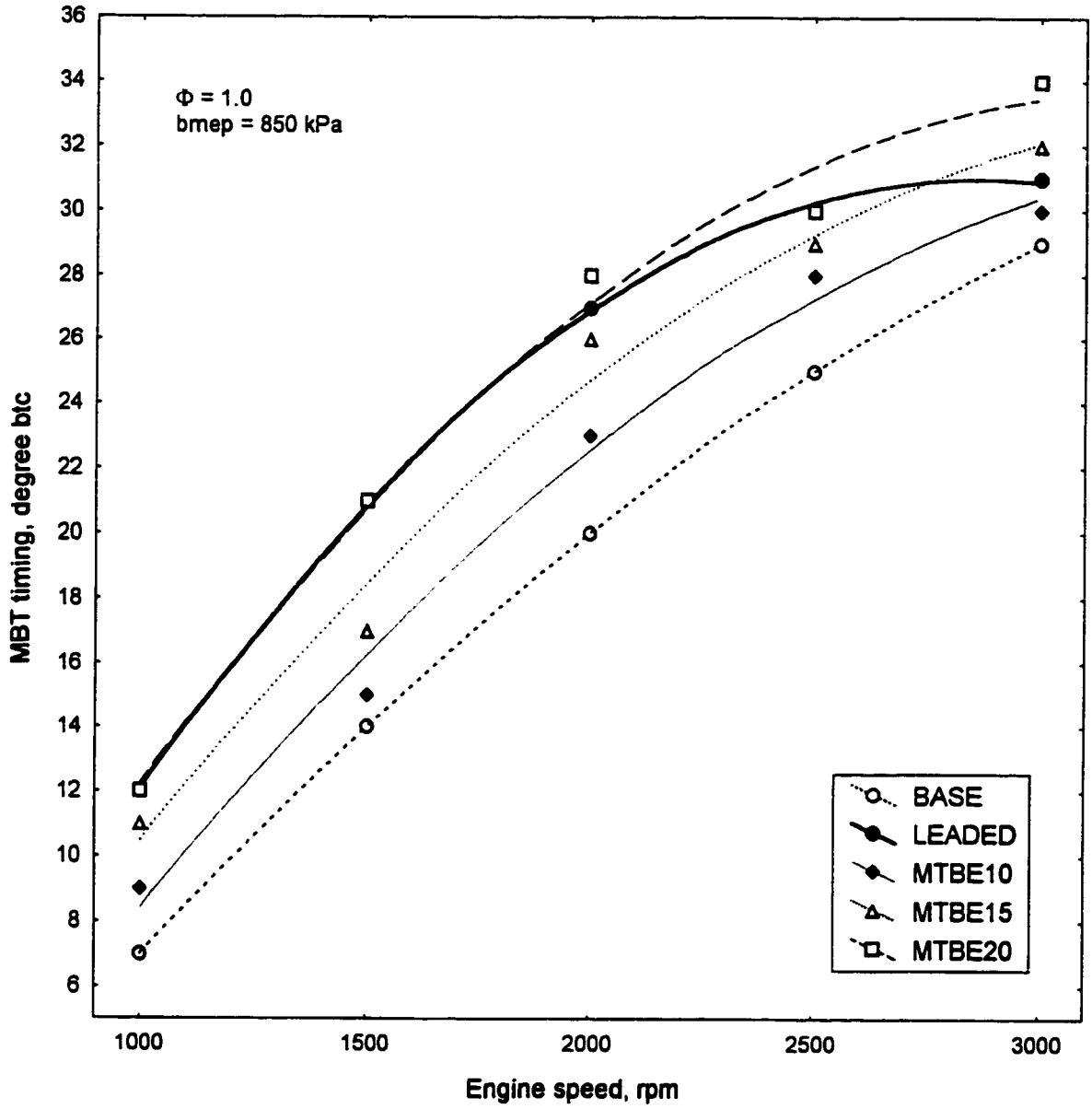

Figure 5-10: MBT timing vs. speed at a constant load of 200 Nm (bmep = 850 kPa) for the MTBE blends.

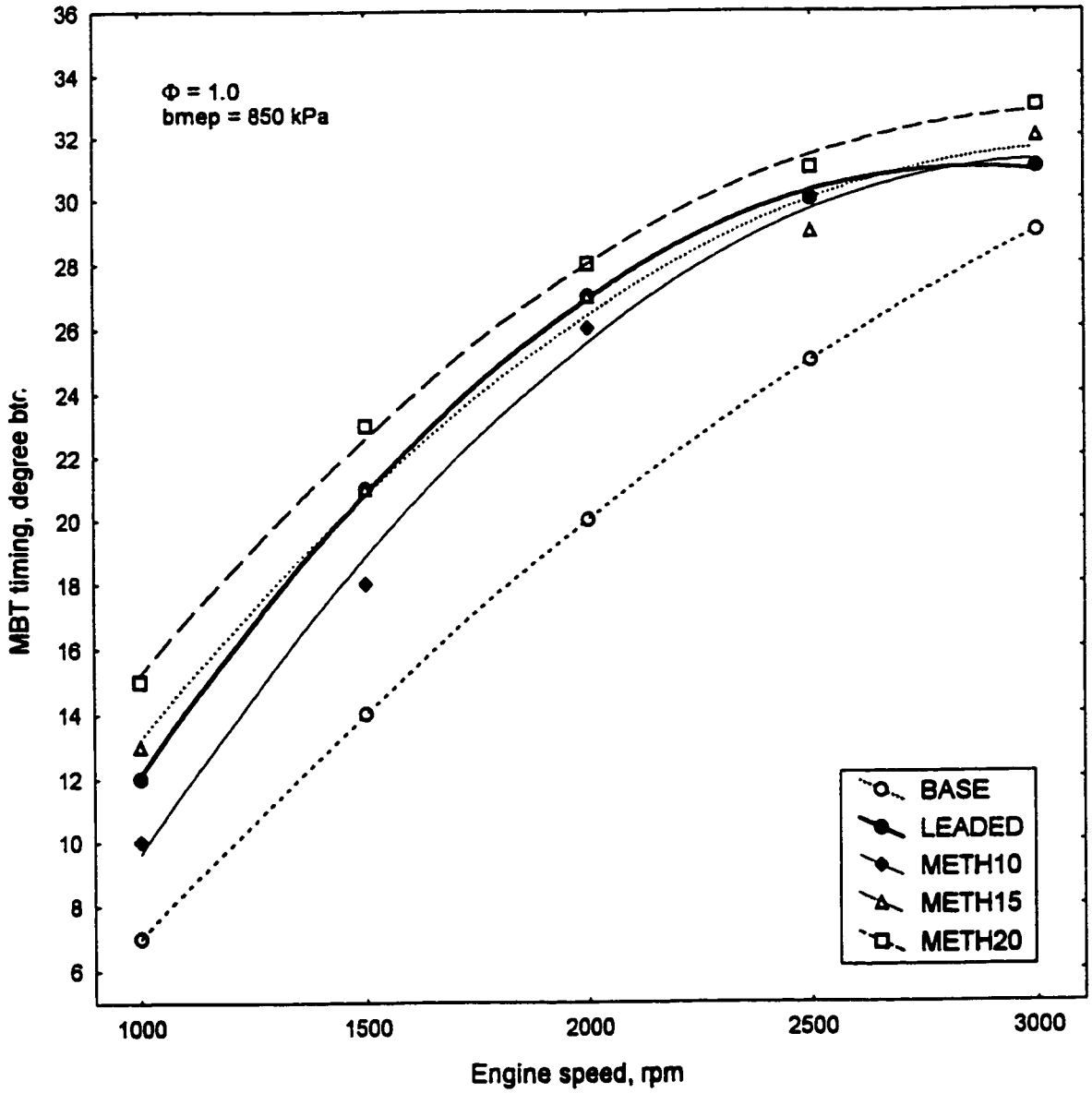

Figure 5-11: MBT timing vs. speed at a constant load of 200 Nm (bmep = 850 kPa) for the methanol blends.

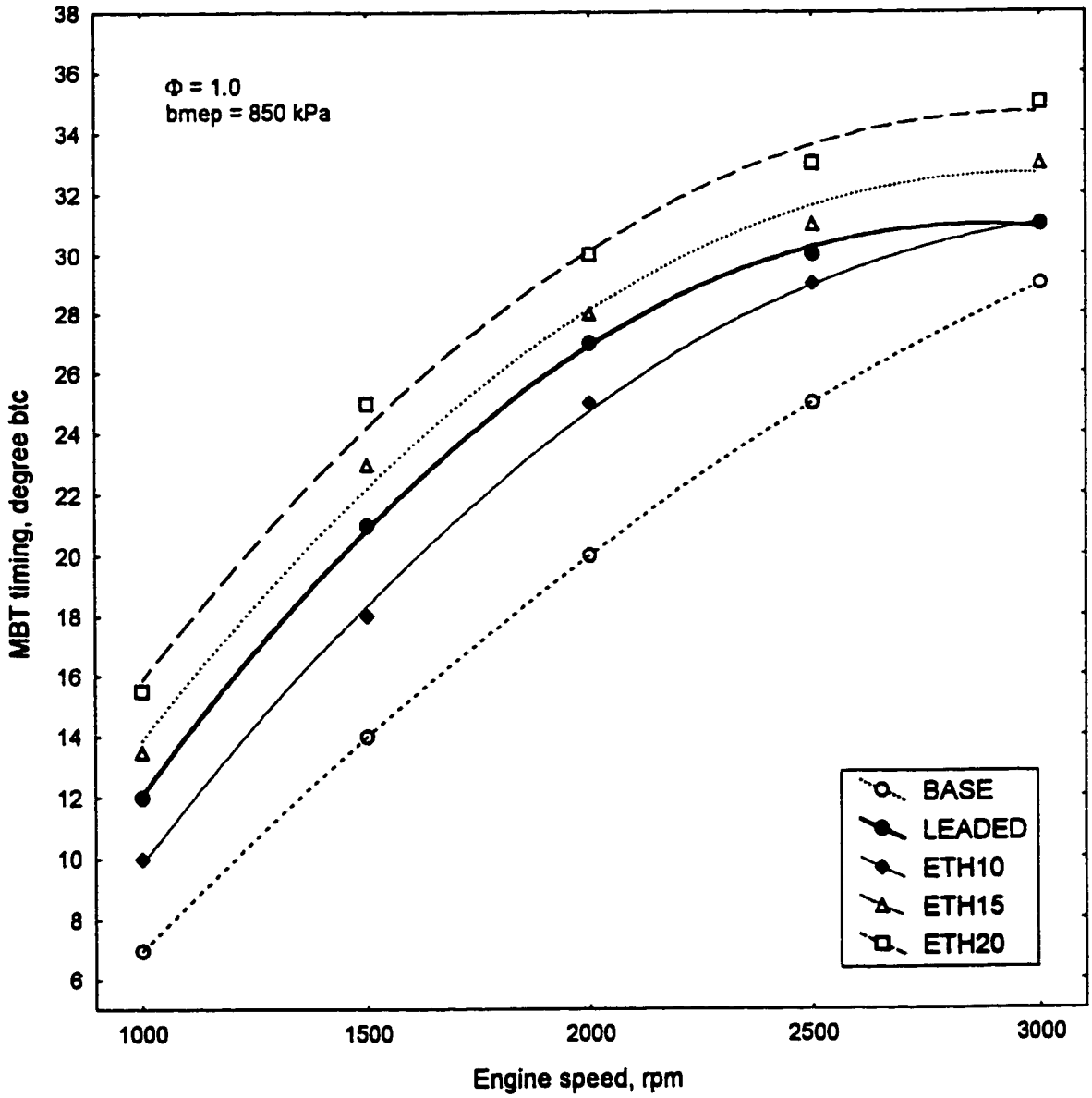

Figure 5-12: MBT timing vs. speed at a constant load of 200 Nm (bmep = 850 kPa) for the ethanol blends.

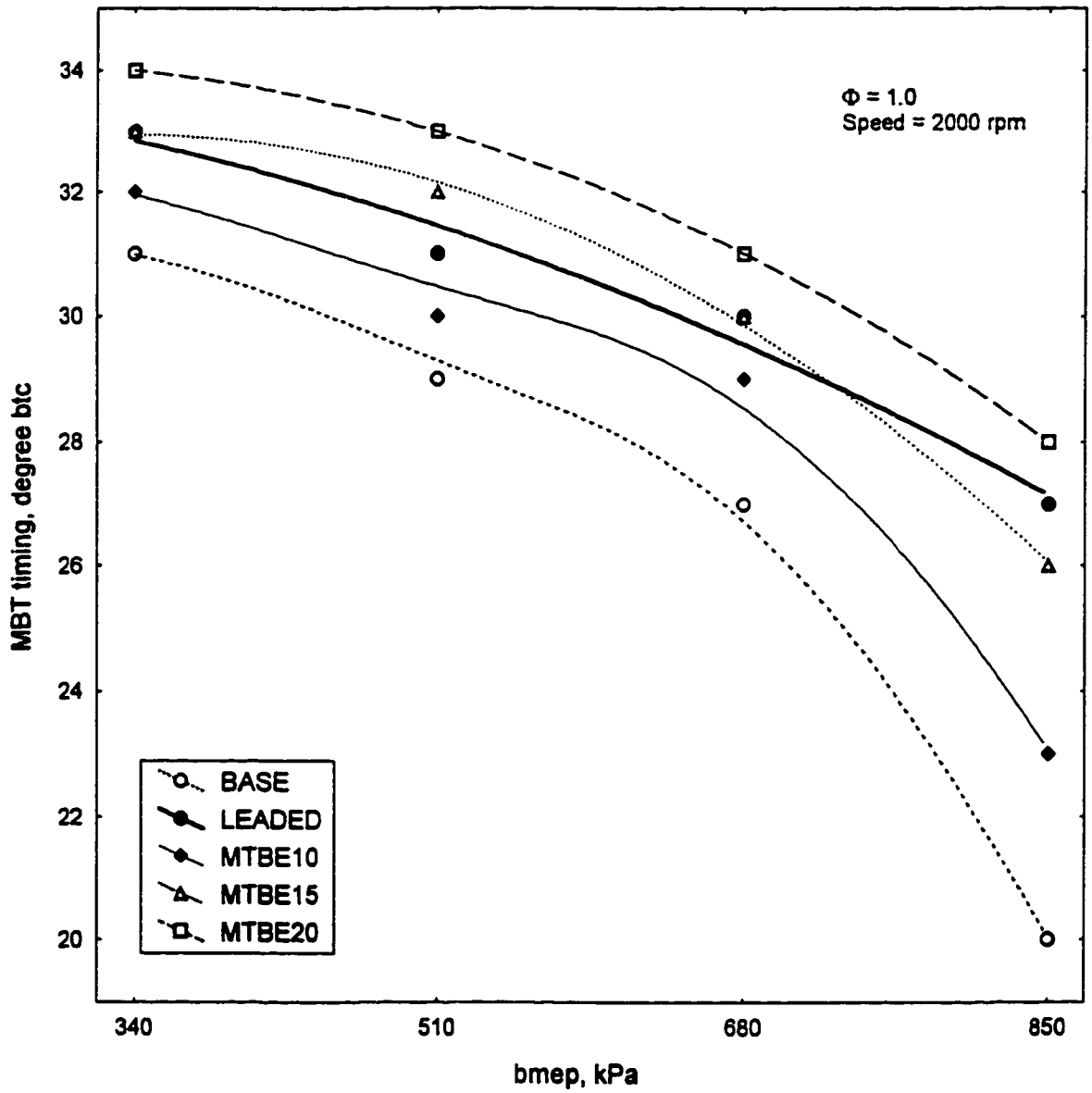

Figure 5-13: Effect of load on MBT timing at a constant speed of 2000 rpm for the MTBE blends.

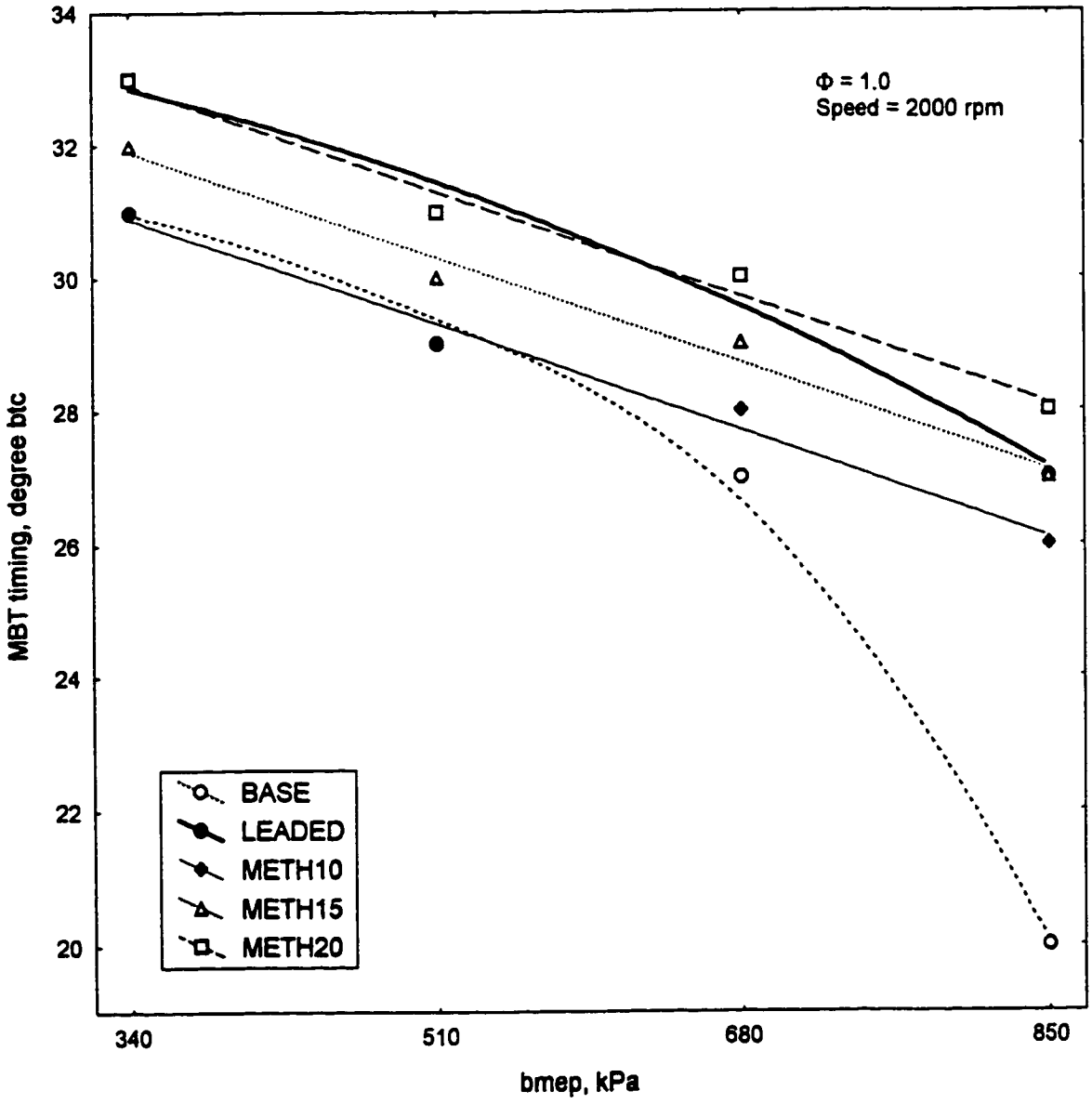

Figure 5-14: Effect of load on MBT timing at a constant speed of 2000 rpm for the methanol blends.

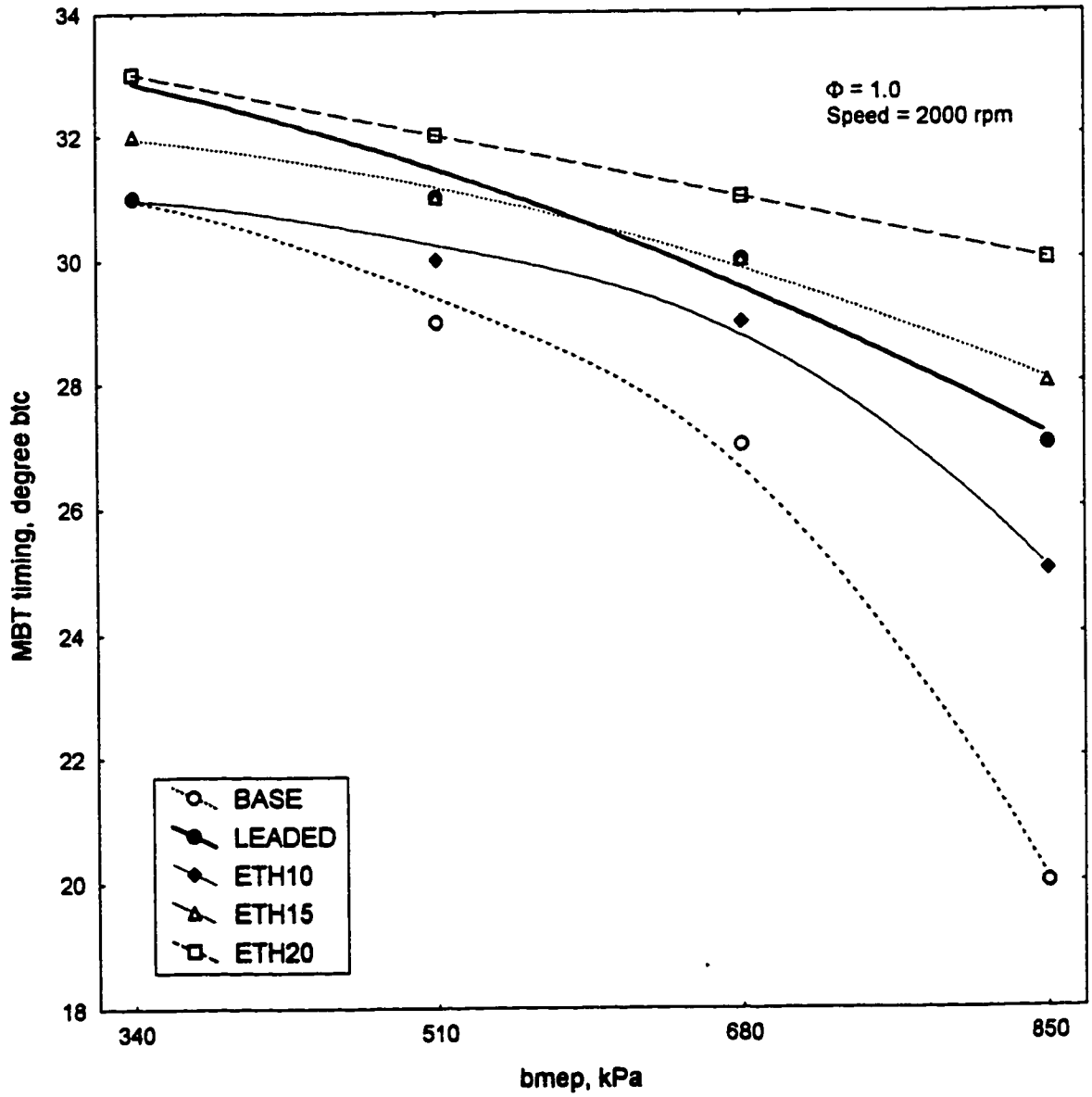

Figure 5-15: Effect of load on MBT timing at a constant speed of 2000 rpm for the ethanol blends.

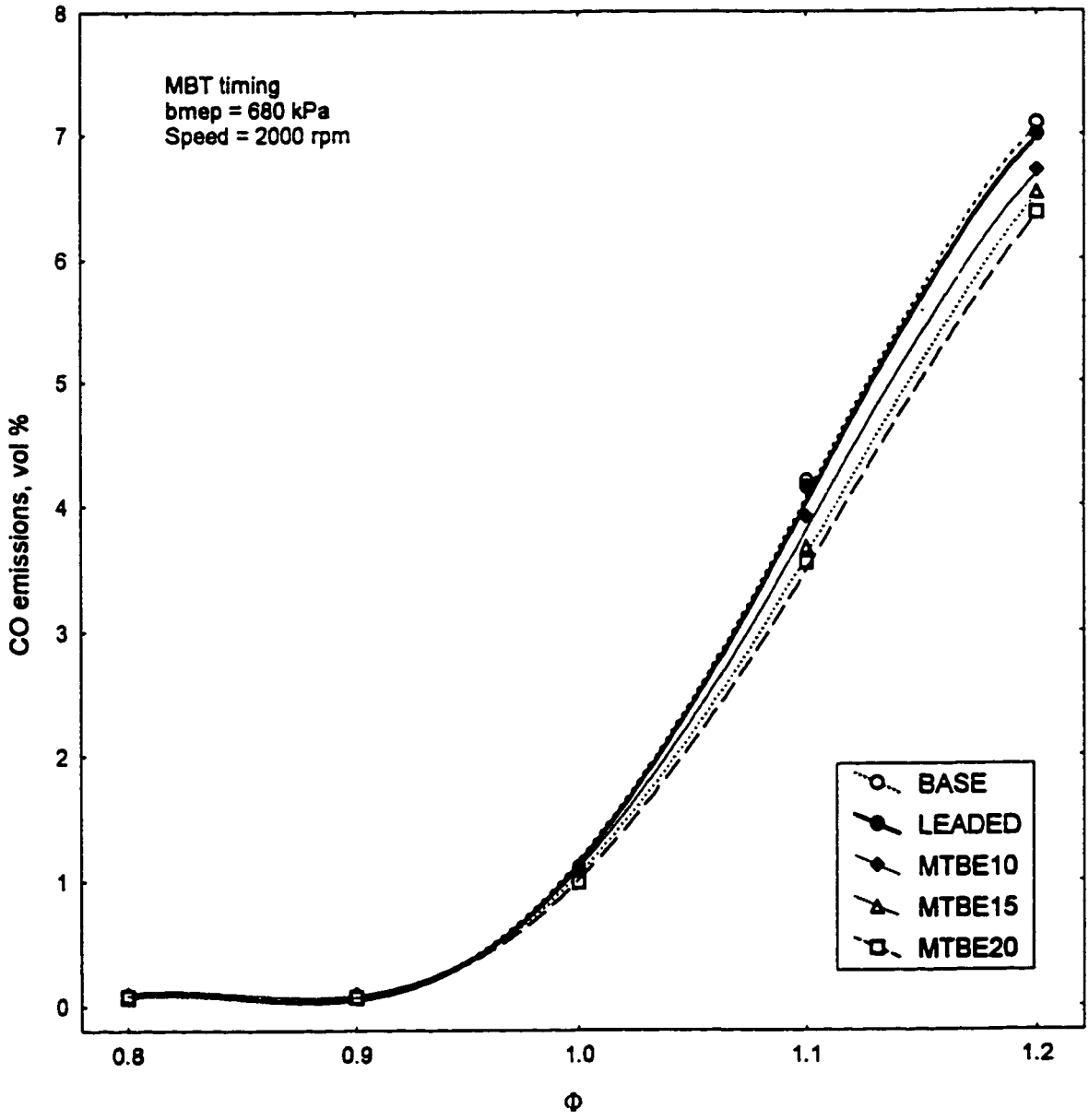

Figure 5-16: Effect of equivalence ratio on CO emissions for the MTBE blends.

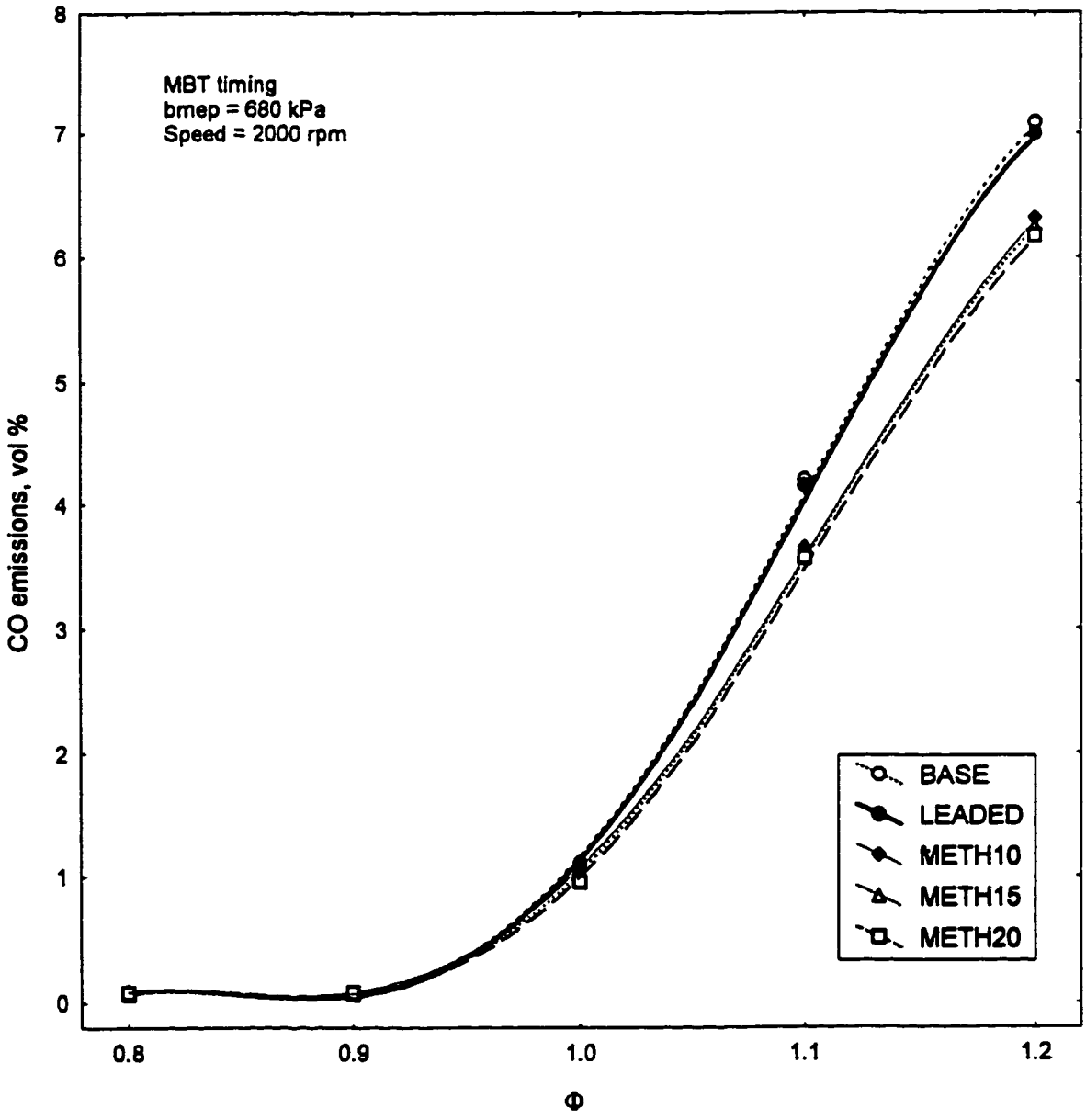

Figure 5-17: Effect of equivalence ratio on CO emissions for the methanol blends.

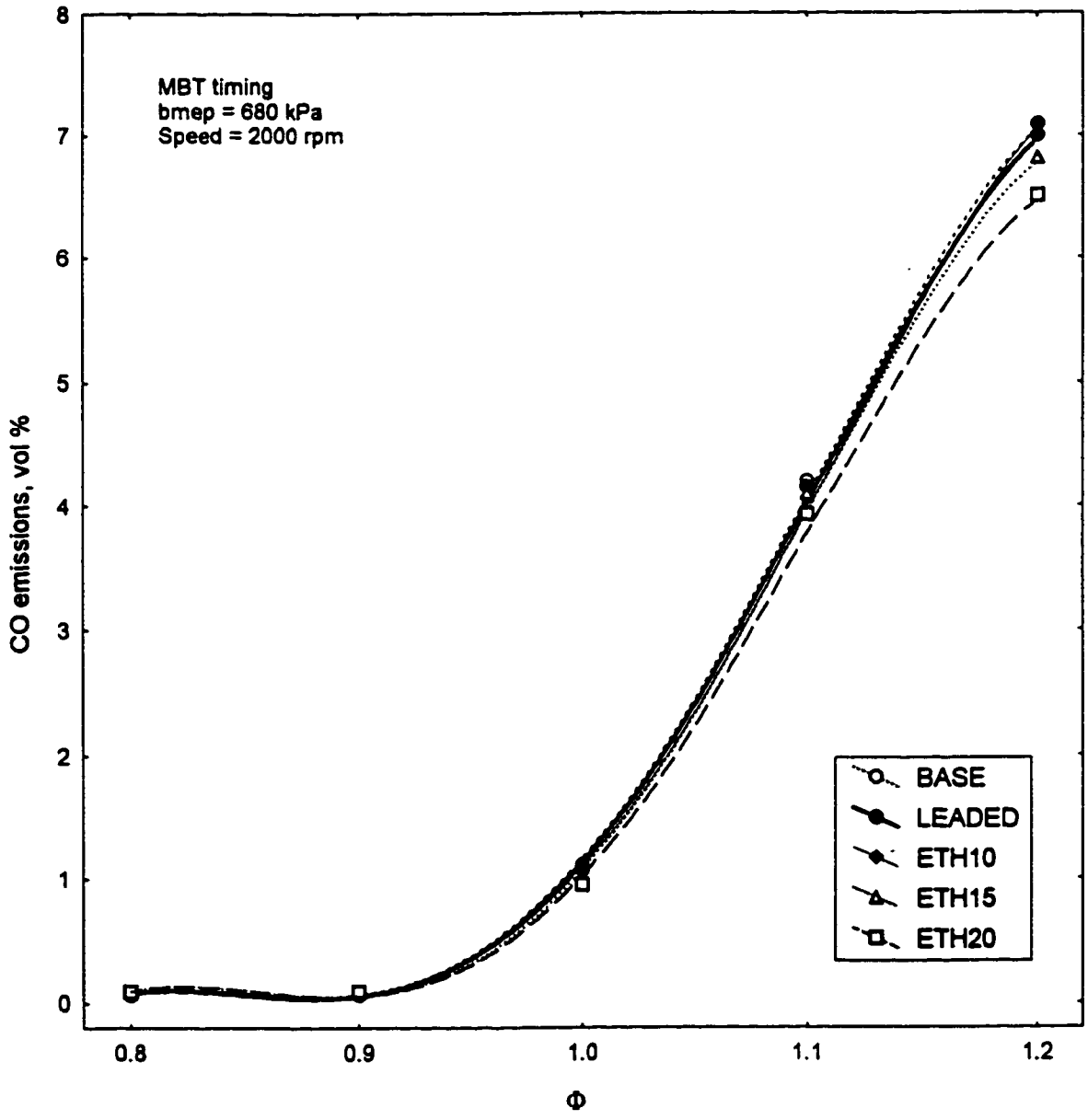

Figure 5-18: Effect of equivalence ratio on CO emissions for the ethanol blends.

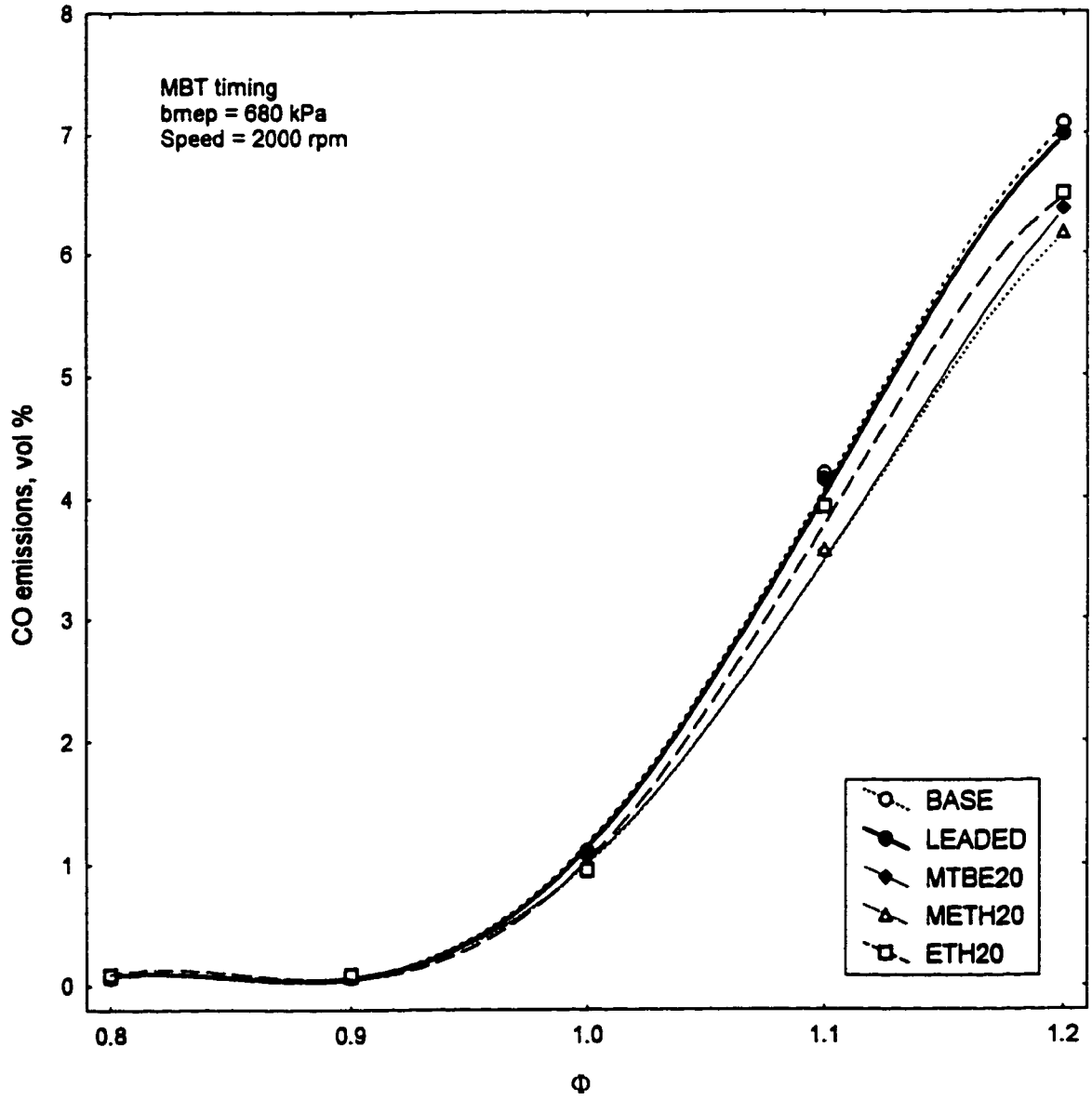

Figure 5-19: Effect of equivalence ratio on CO emissions for the least emitting blends compared to the base and leaded fuels.

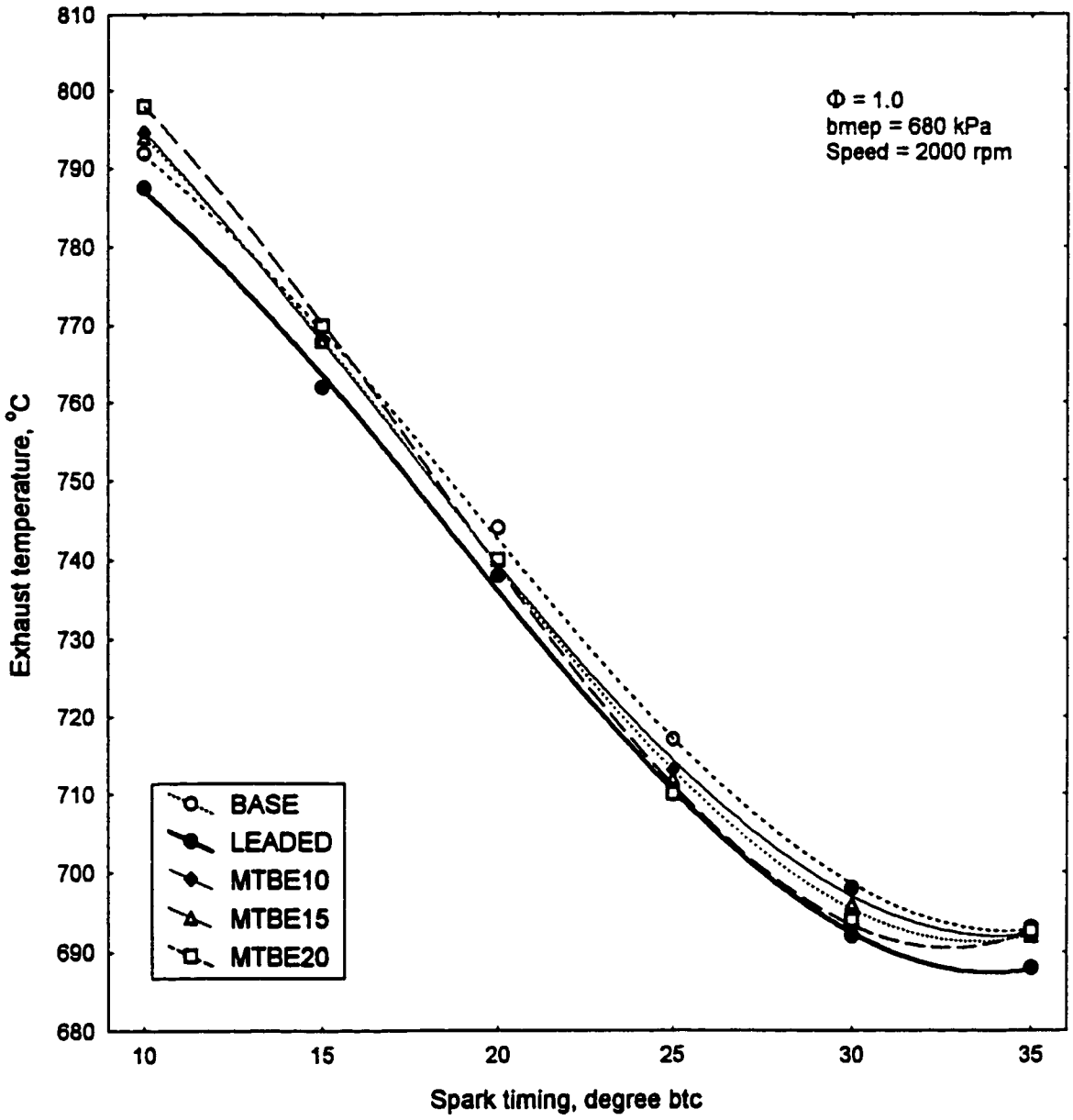

Figure 5-20: Effect of spark timing on exhaust gas temperature for the MTBE blends.

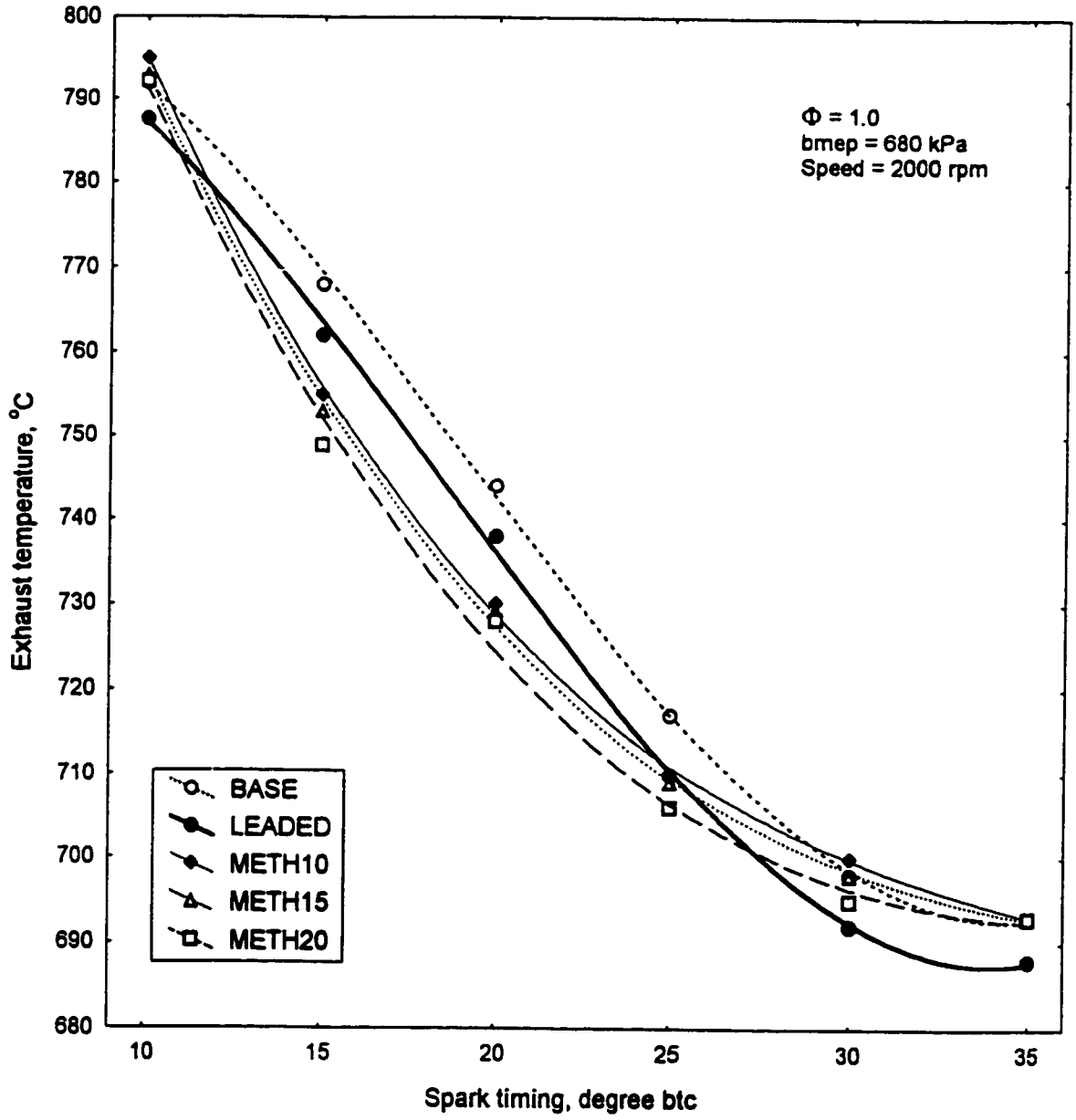

Figure 5-21: Effect of spark timing on exhaust gas temperature for the methanol blends.

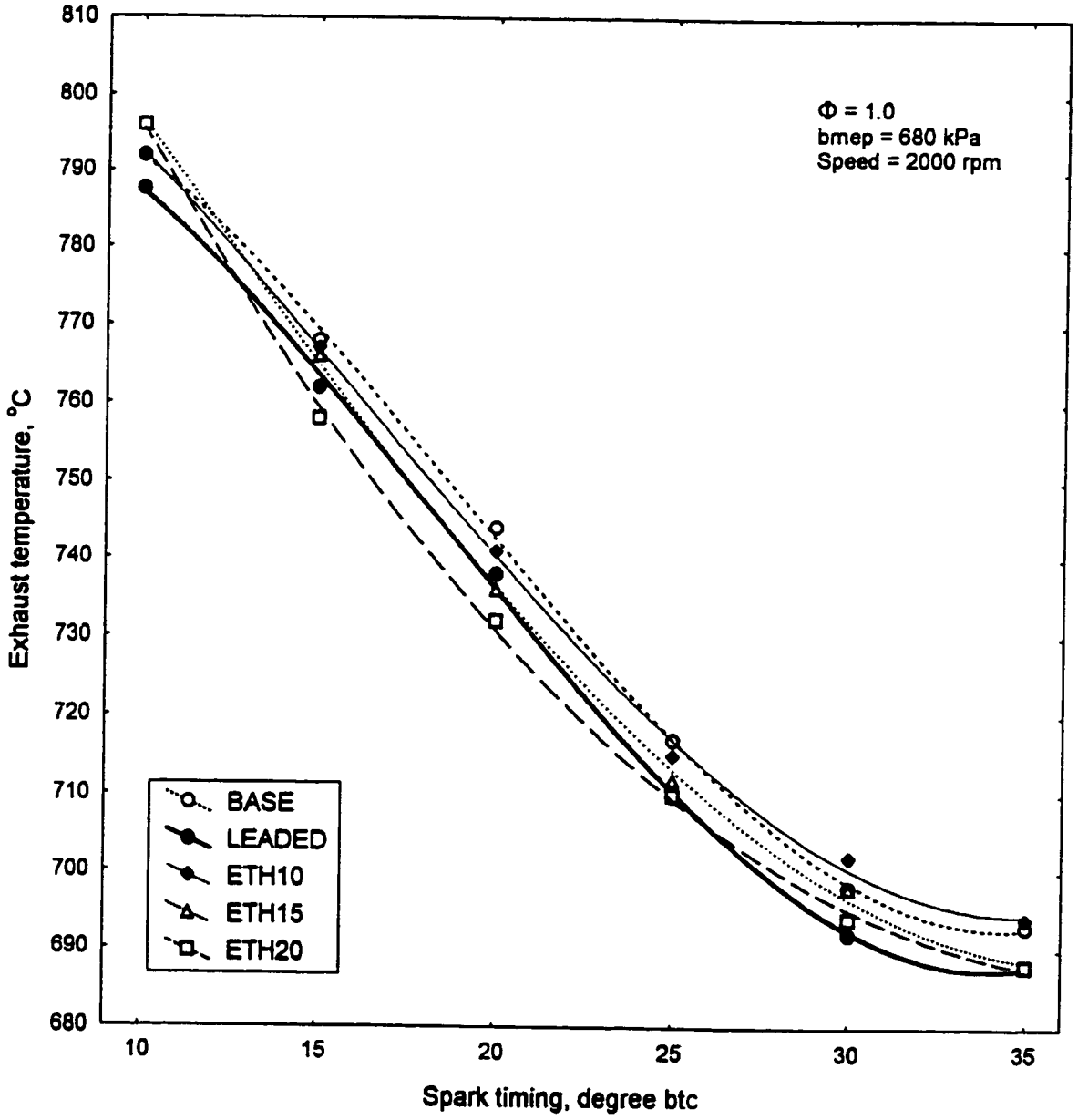

Figure 5-22: Effect of spark timing on exhaust gas temperature for the ethanol blends.

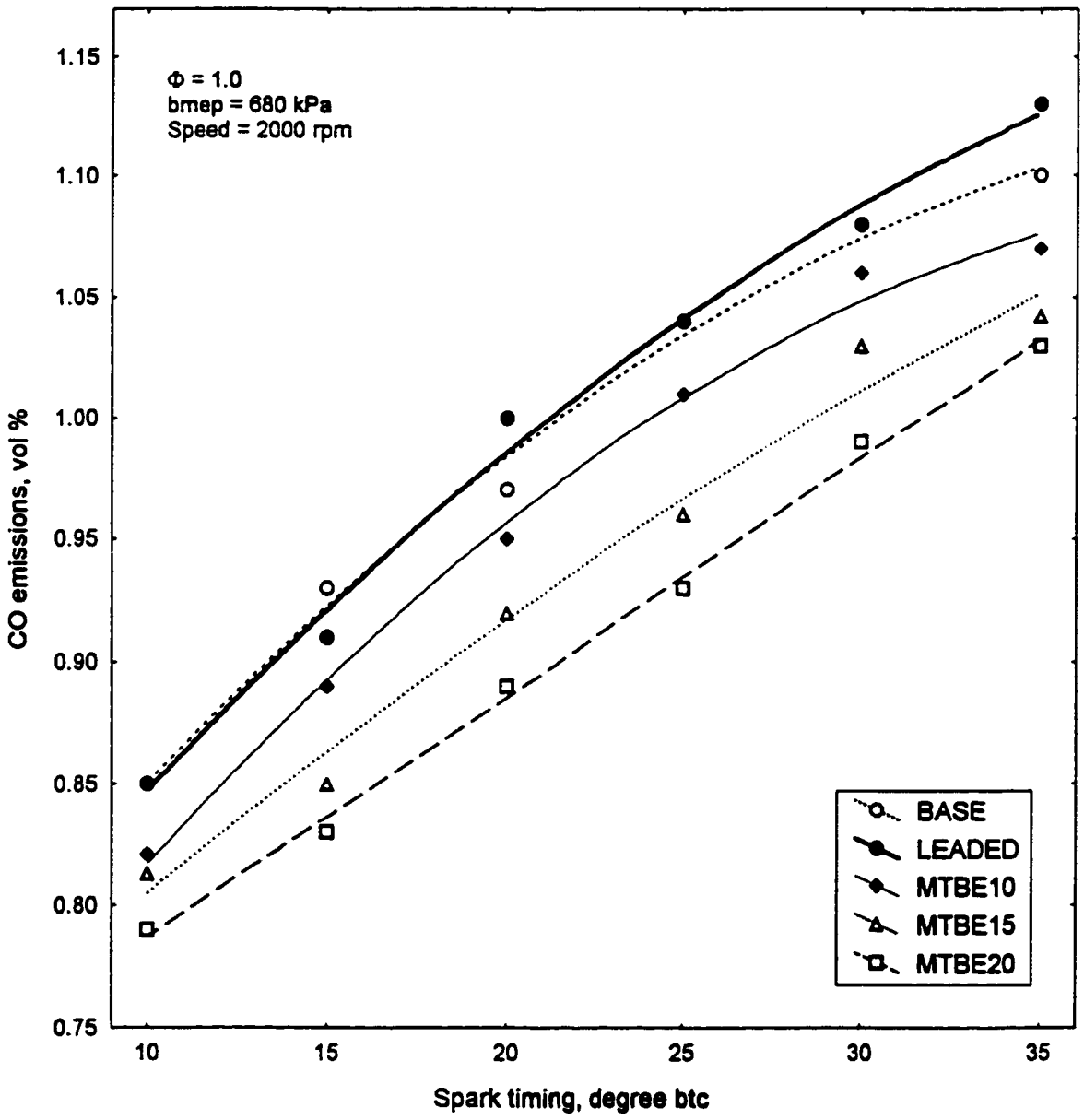

Figure 5-23: Effect of spark timing on CO emissions for the MTBE blends.

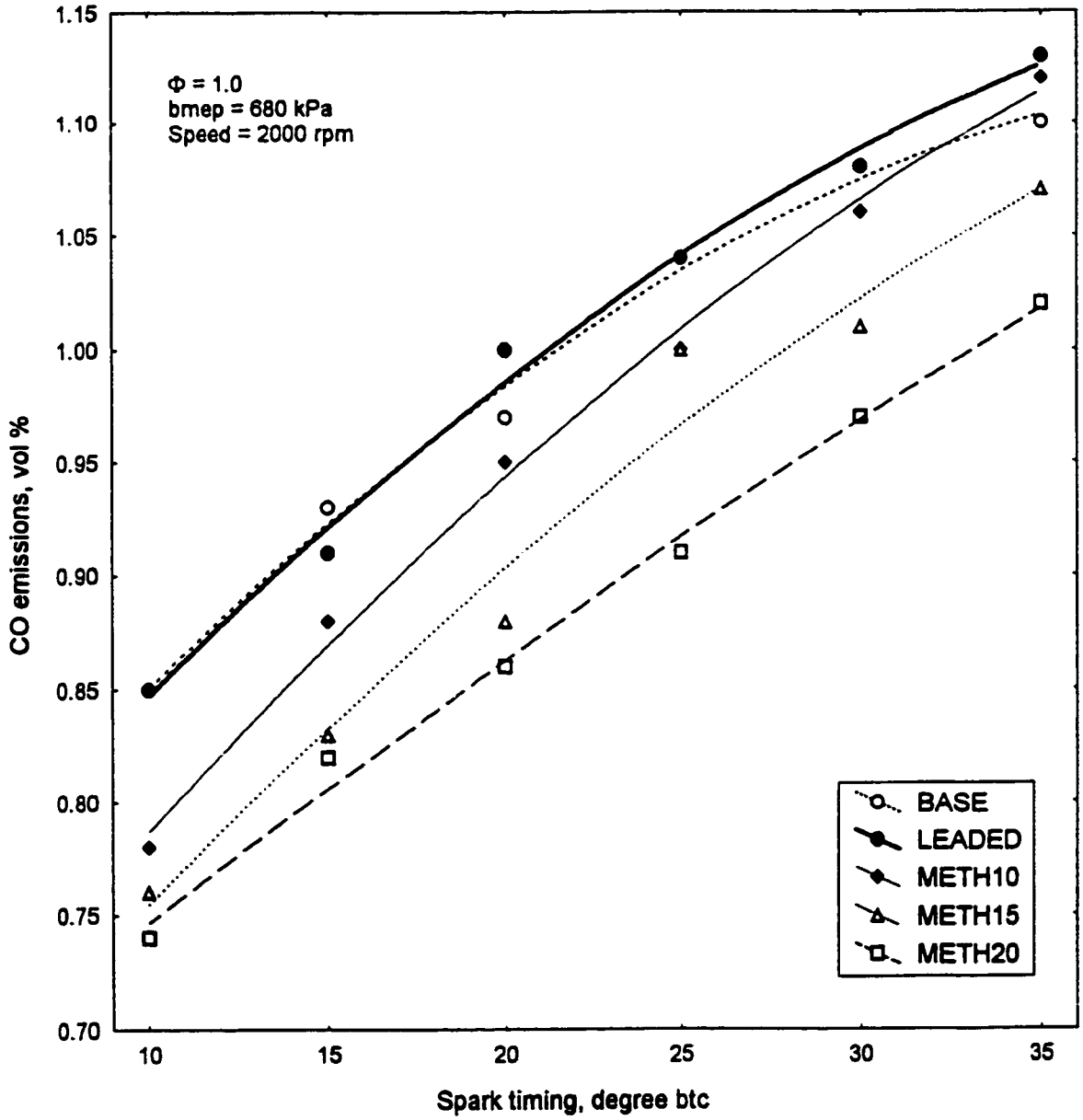

Figure 5-24: Effect of spark timing on CO emissions for the methanol blends.

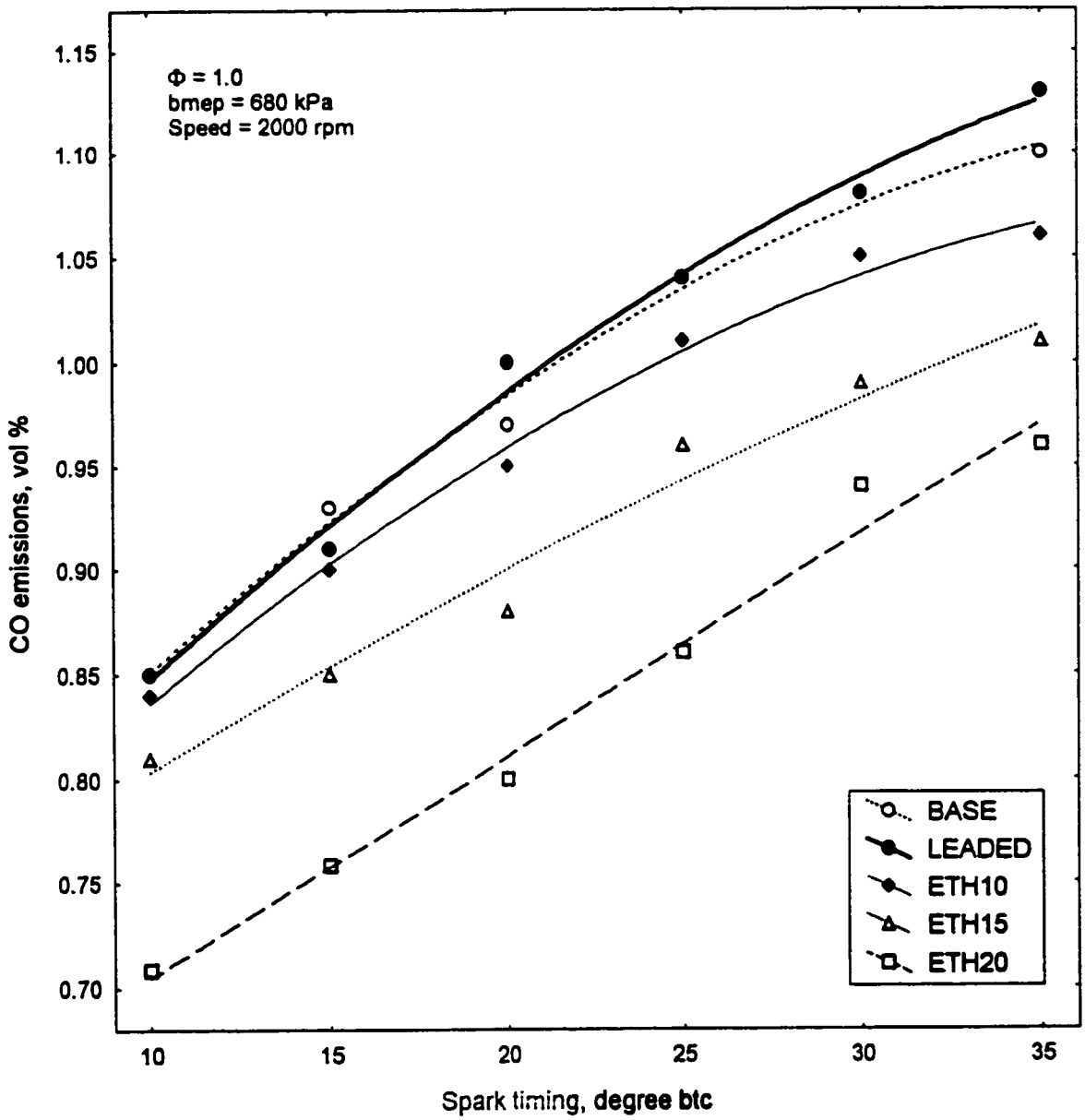

Figure 5-25: Effect of spark timing on CO emissions for the ethanol blends.

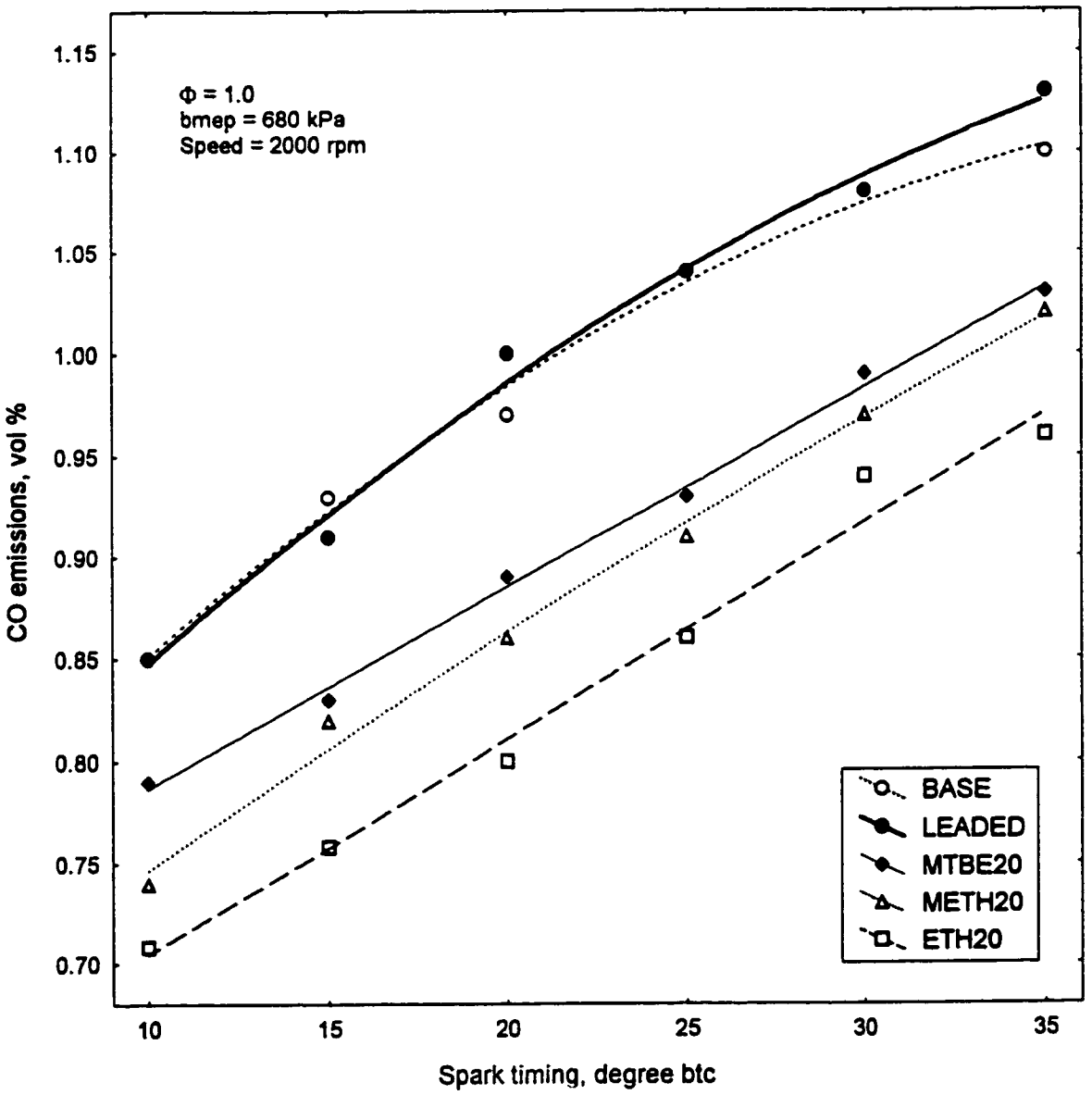

Figure 5-26: Effect of spark timing on CO emissions for the least emitting blends compared to the base and leaded fuels.

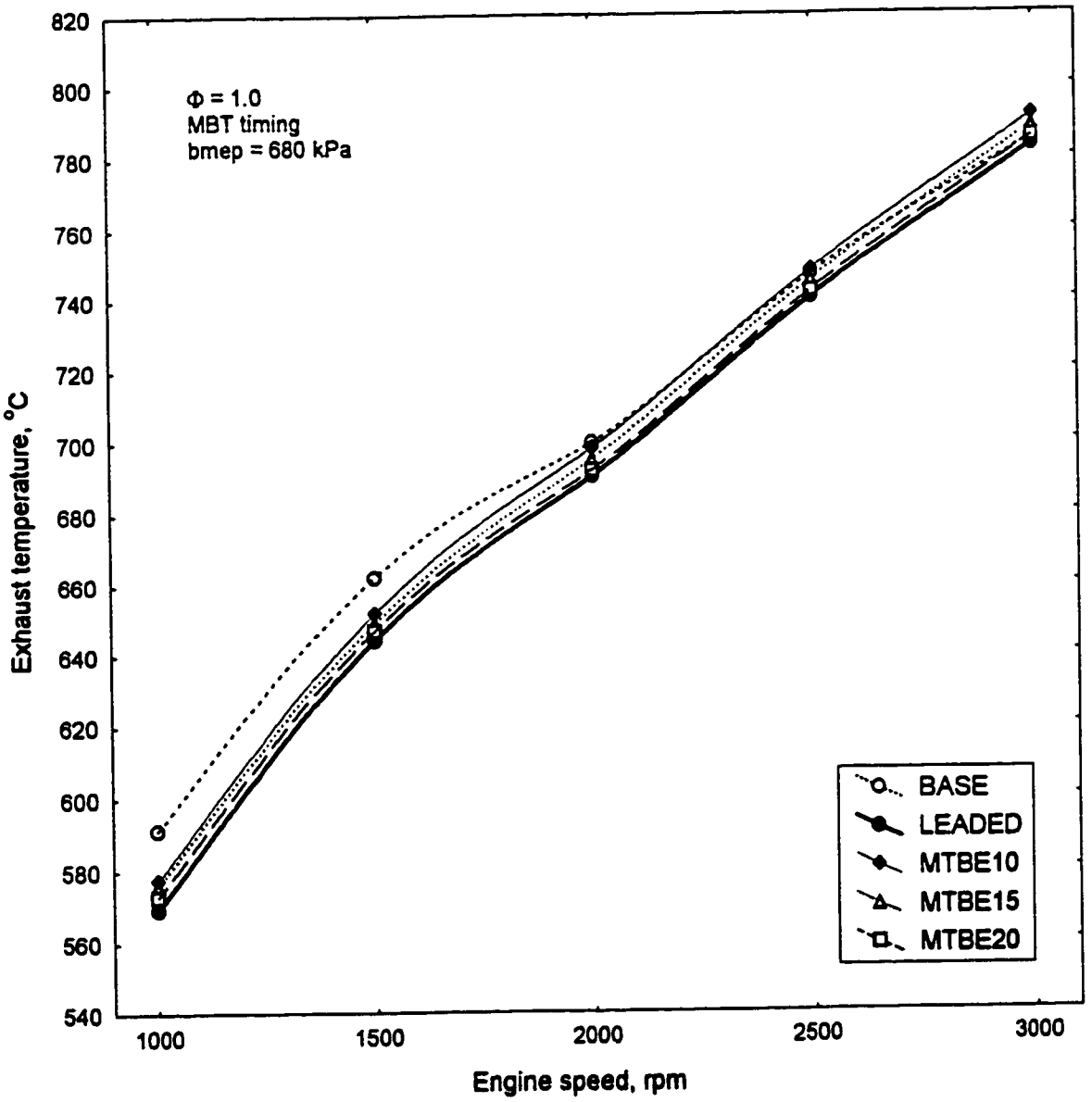

Figure 5-27: Exhaust gas temperature vs. speed at a constant load of 160 Nm (bmep = 680 kPa) for the MTBE blends.

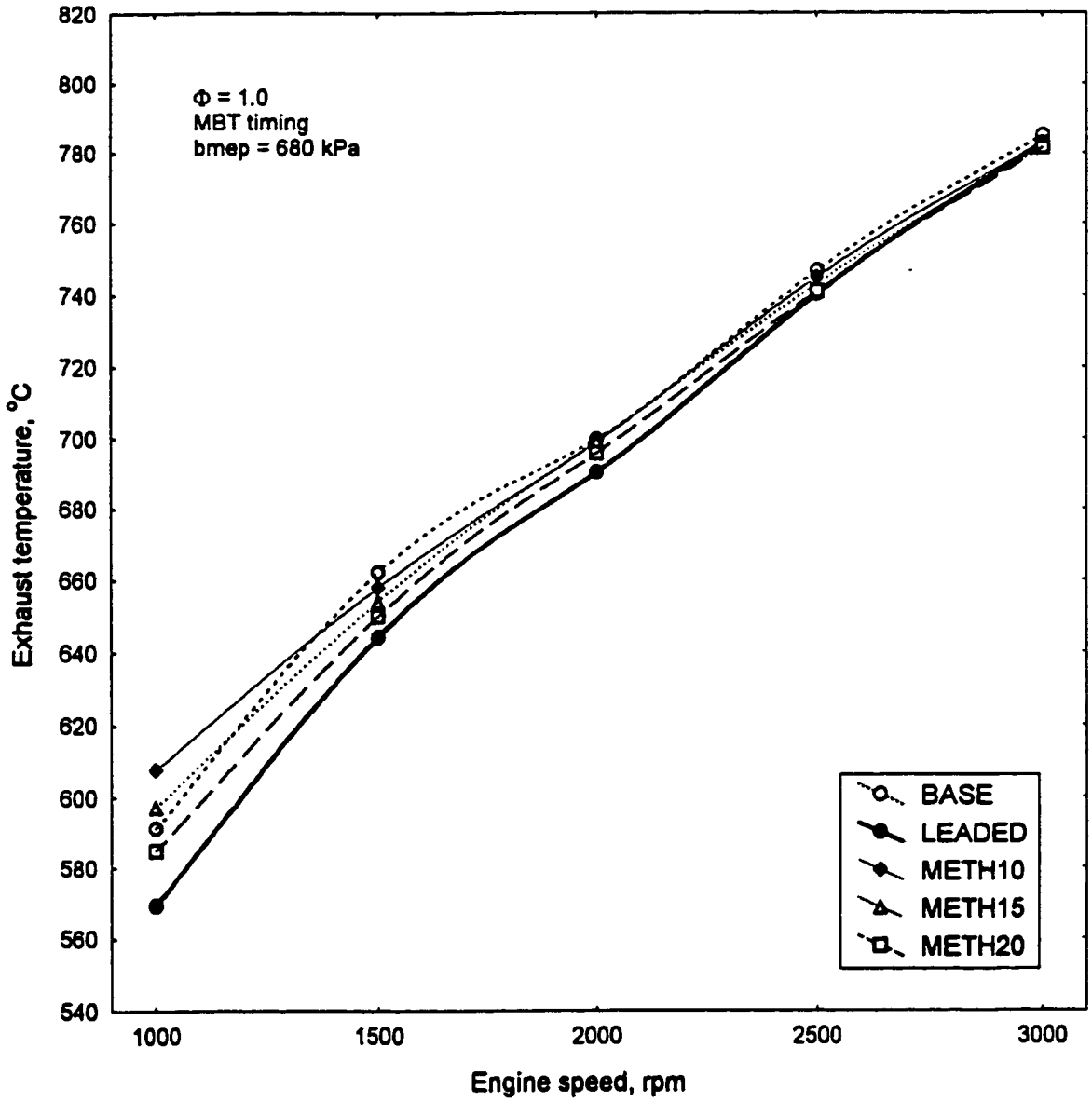

Figure 5-28: Exhaust gas temperature vs. speed at a constant load of 160 Nm (bmep = 680 kPa) for the methanol blends.

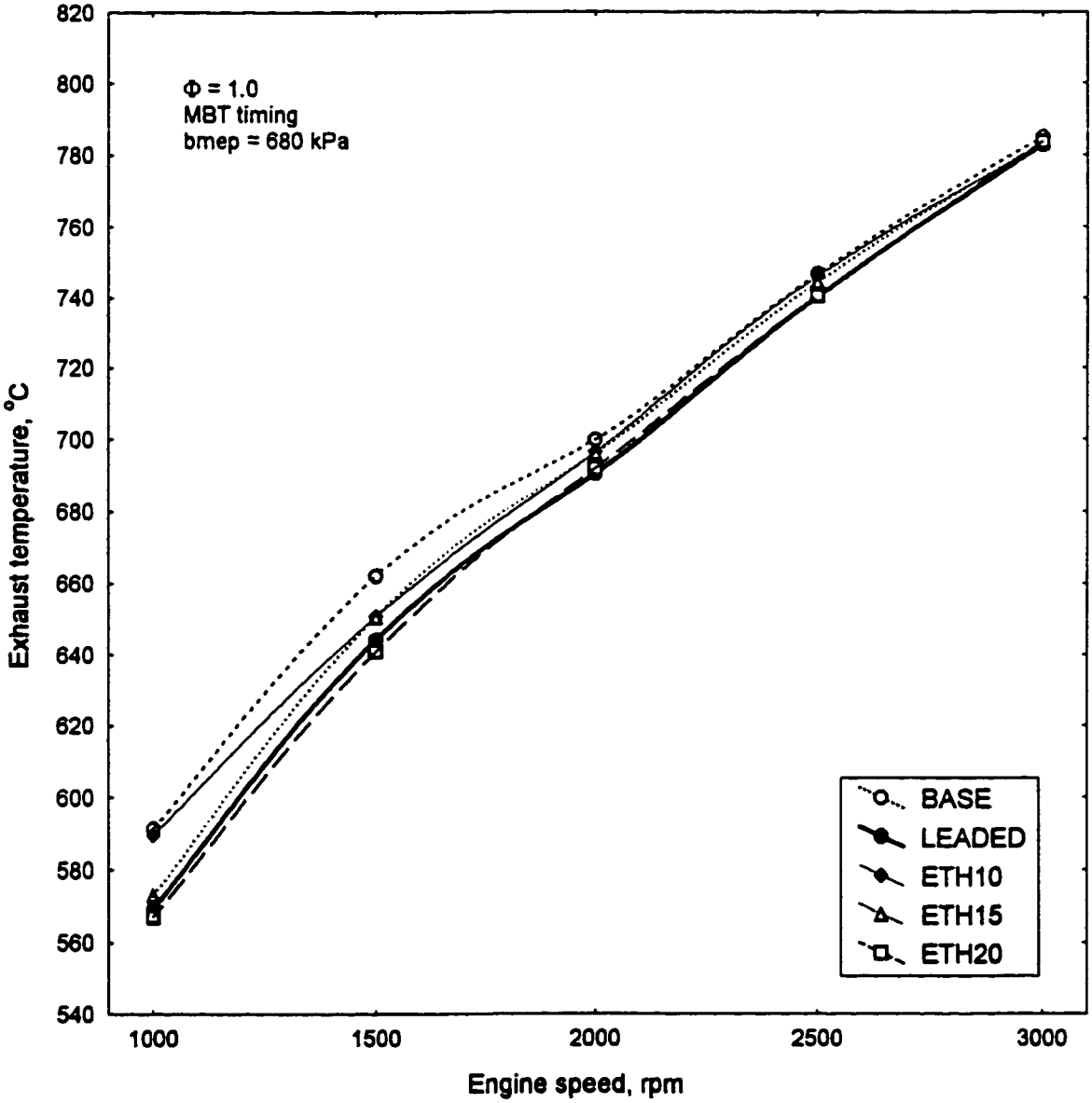

Figure 5-29: Exhaust gas temperature vs. speed at a constant load of 160 Nm (bmep = 680 kPa) for the ethanol blends.

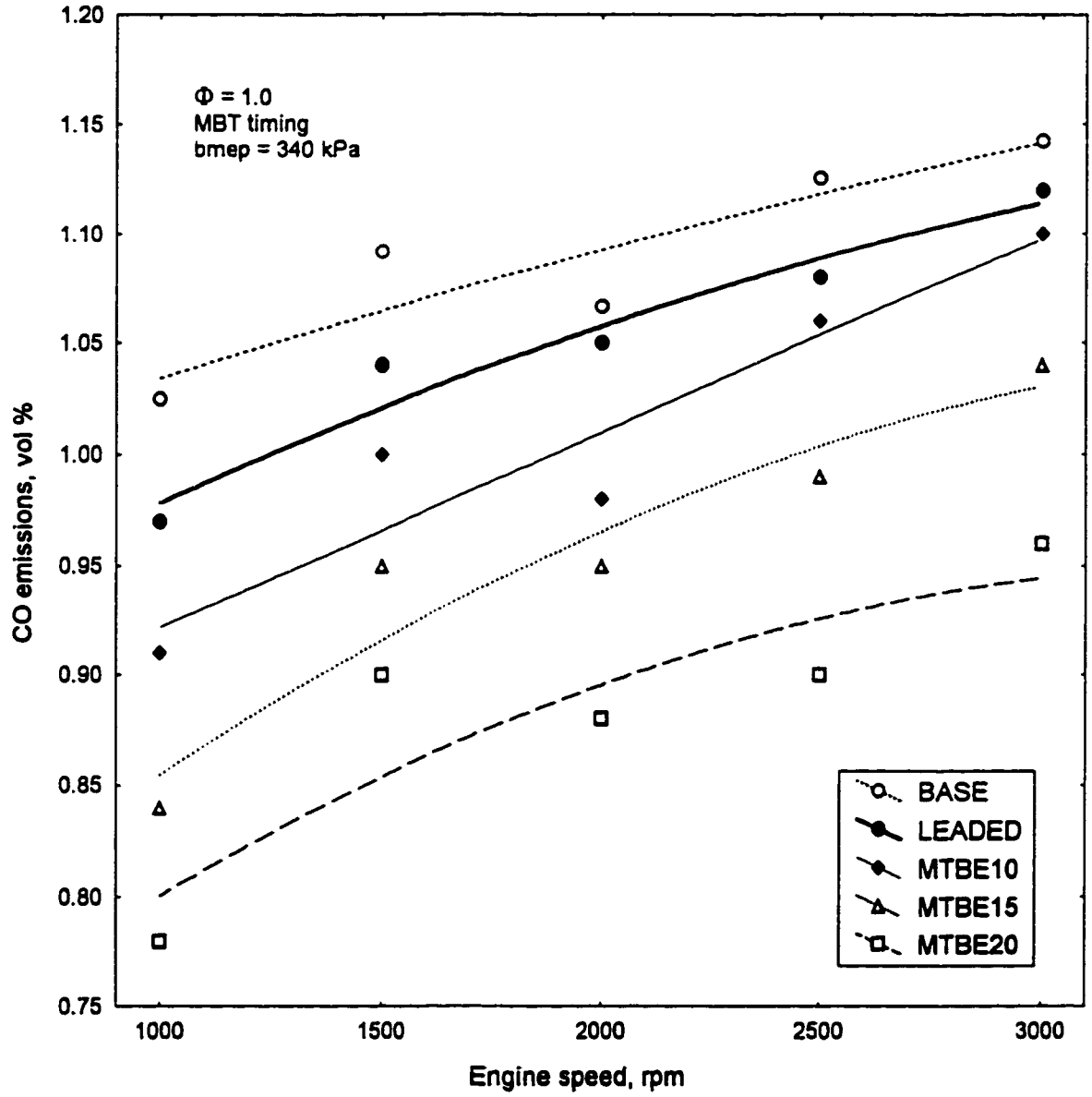

Figure 5-30: CO emissions vs. speed at a constant load of 80 Nm (bmep = 340 kPa) for the MTBE blends.

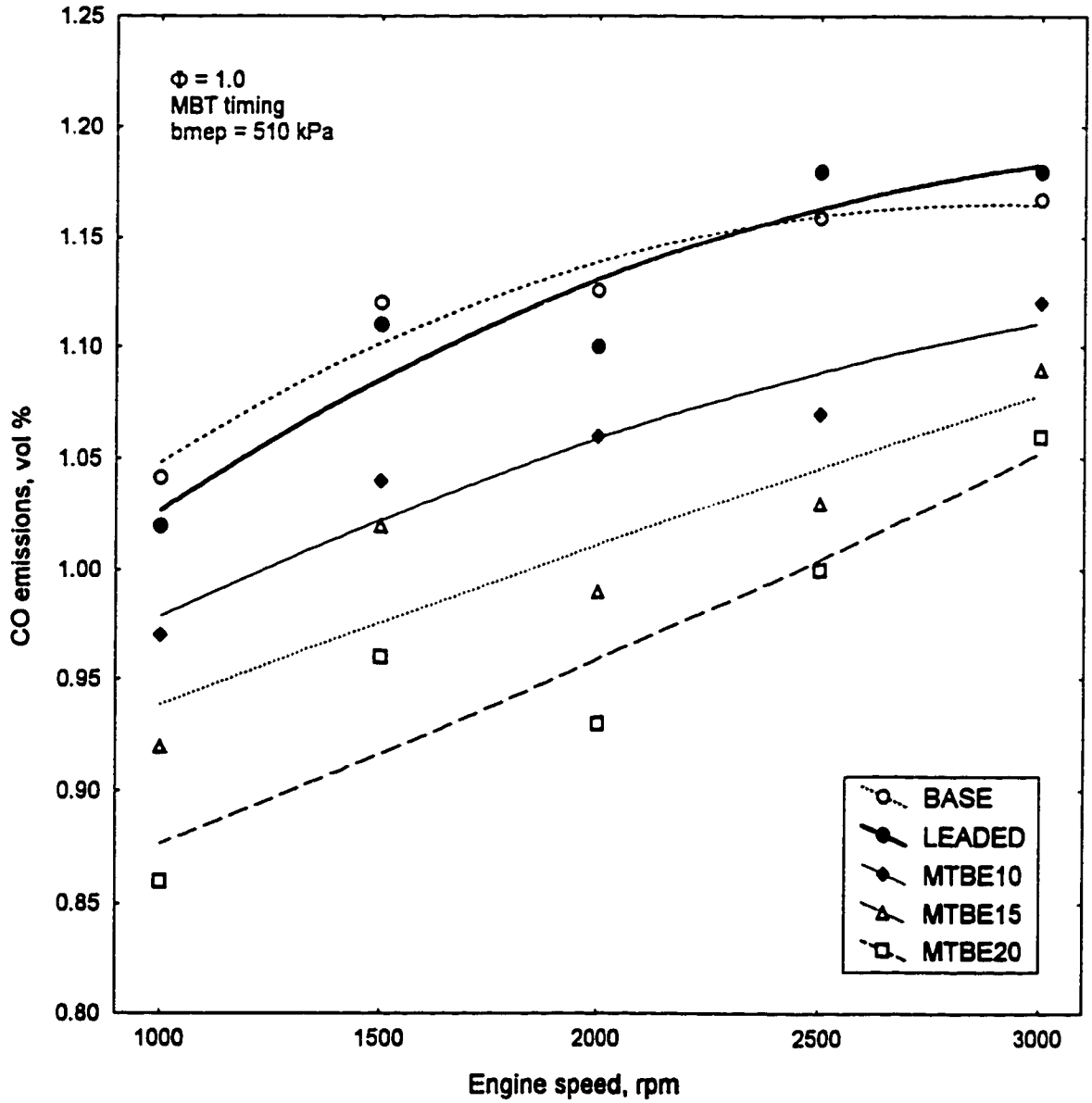

Figure 5-31: CO emissions vs. speed at a constant Load of 120 Nm (bmep = 510 kPa) for the MTBE blends.

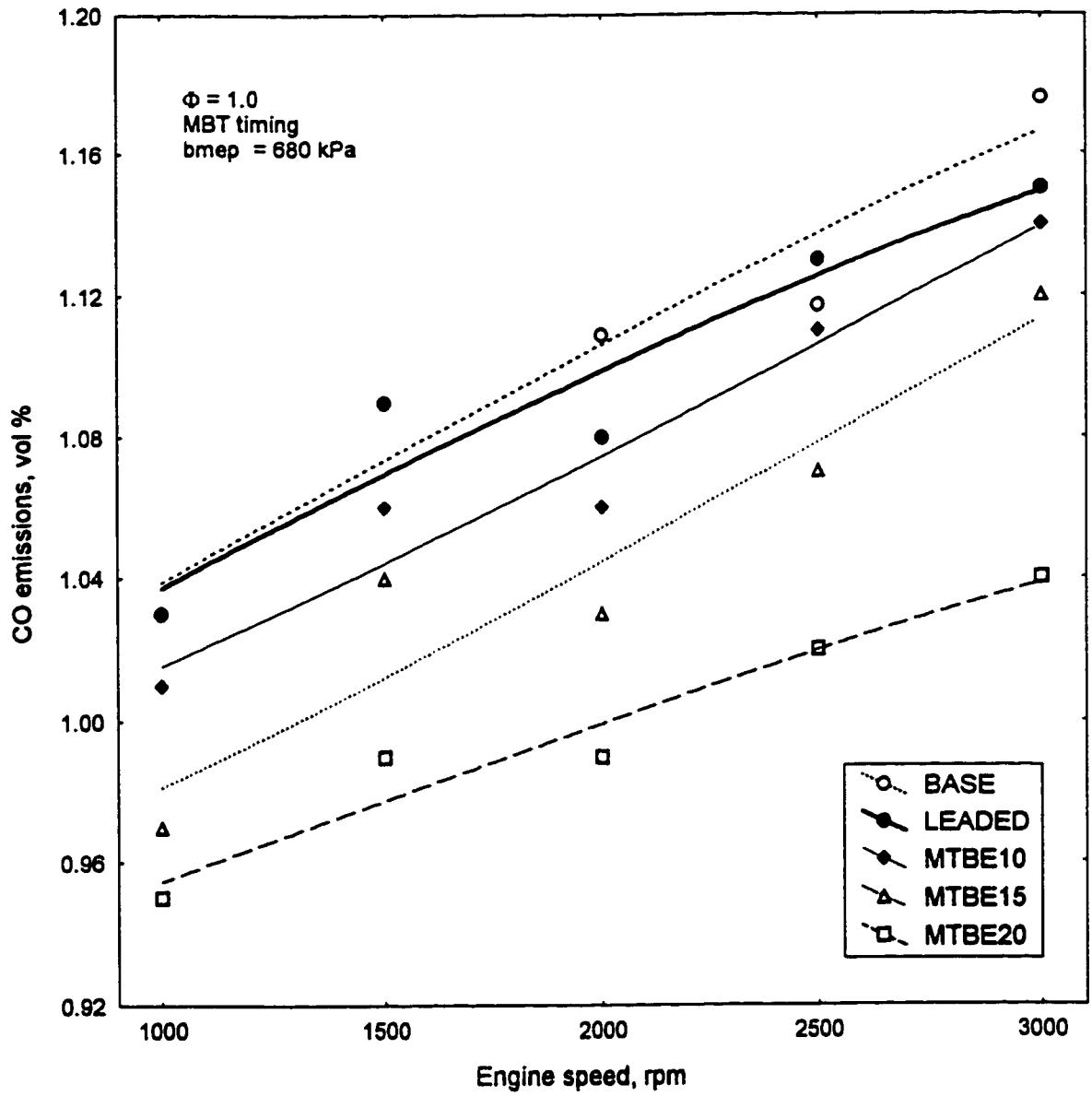

Figure 5-32: CO emissions vs. speed at a constant load of 160 Nm (bmep = 680 kPa) for the MTBE blends.

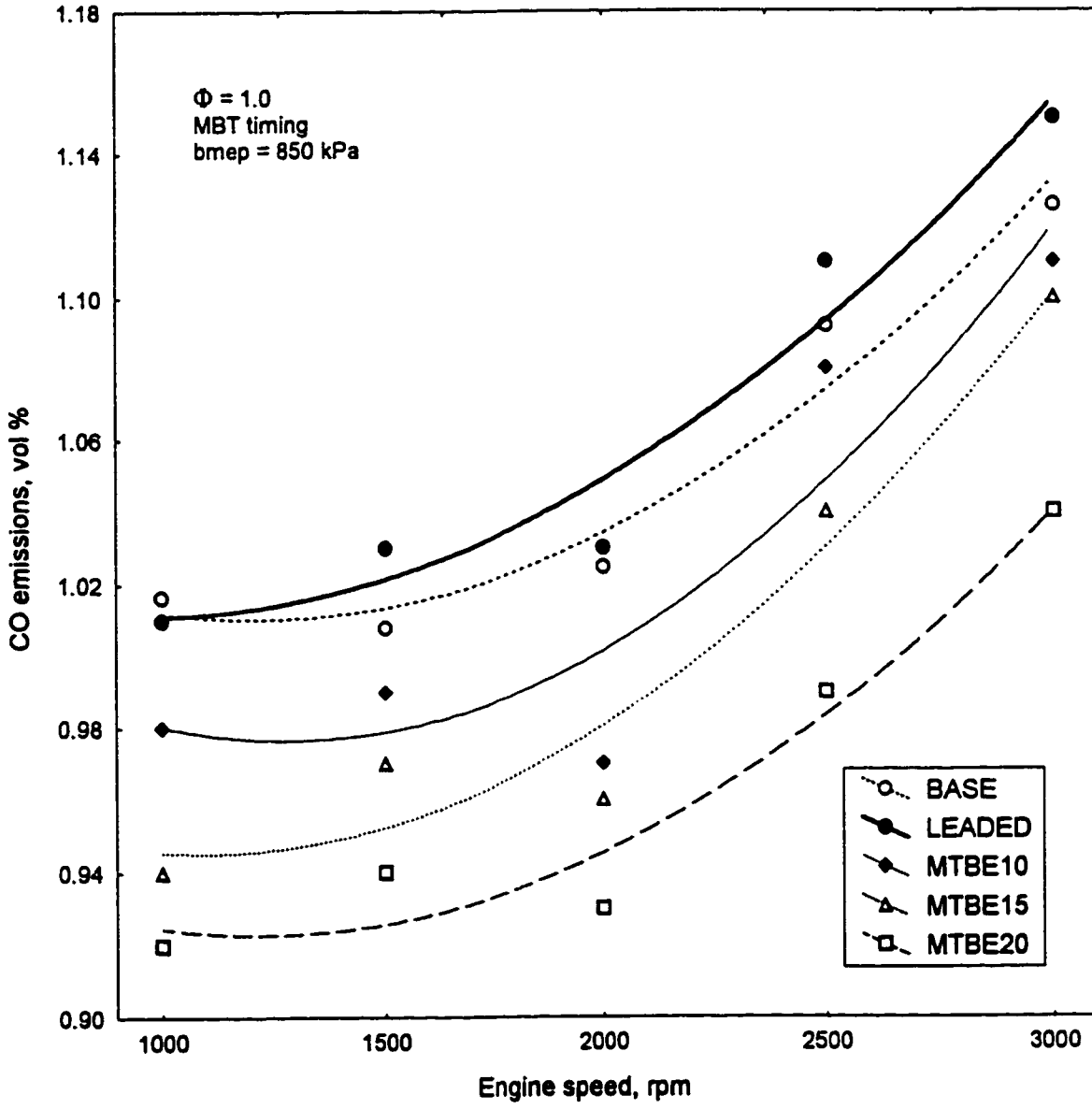

Figure 5-33: CO emissions vs. speed at a constant load of 200 Nm (bmep = 850 kPa) for the MTBE blends.

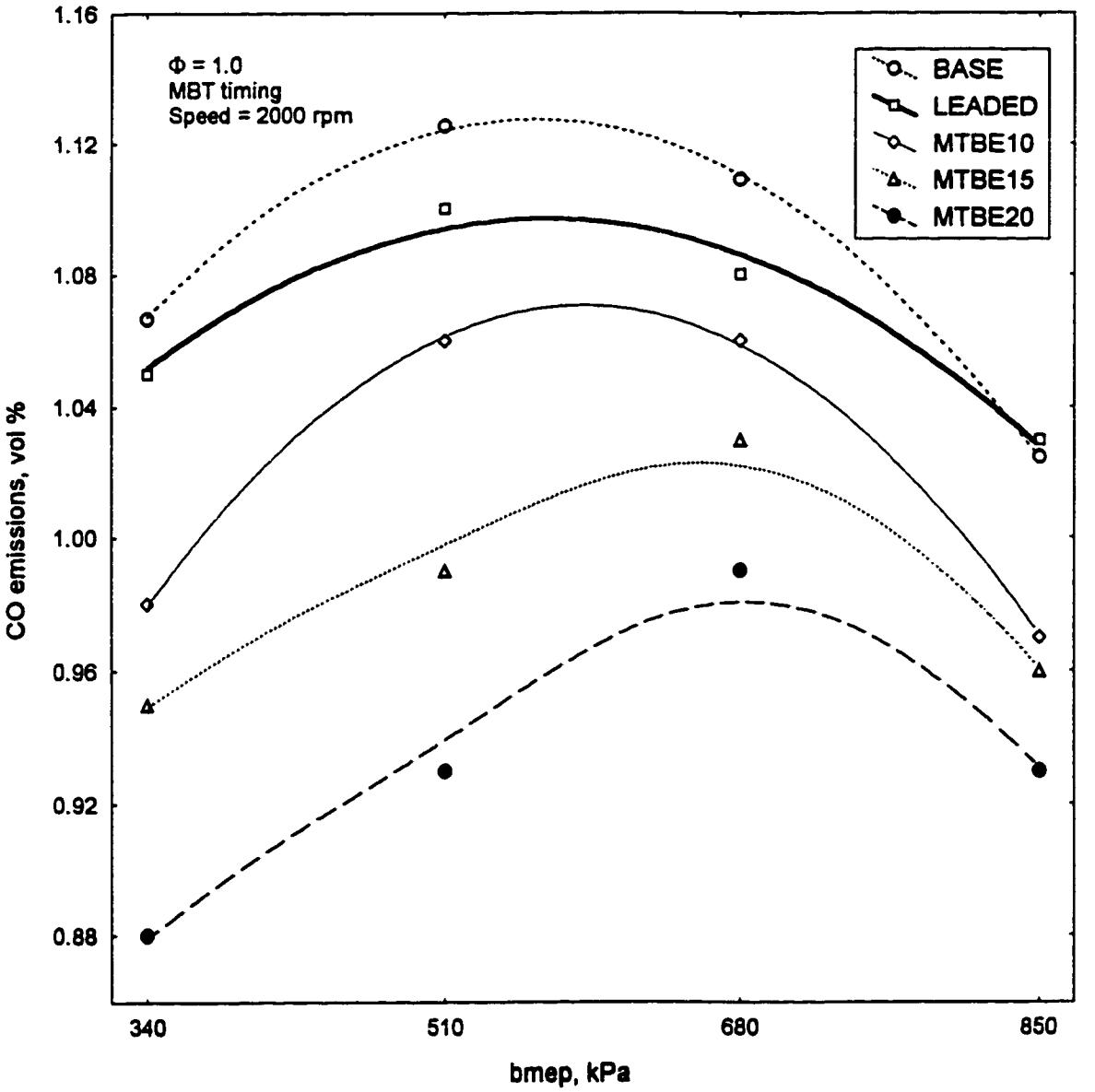

Figure 5-34: Effect of load on CO emissions at a constant speed of 2000 rpm for the MTBE blends.

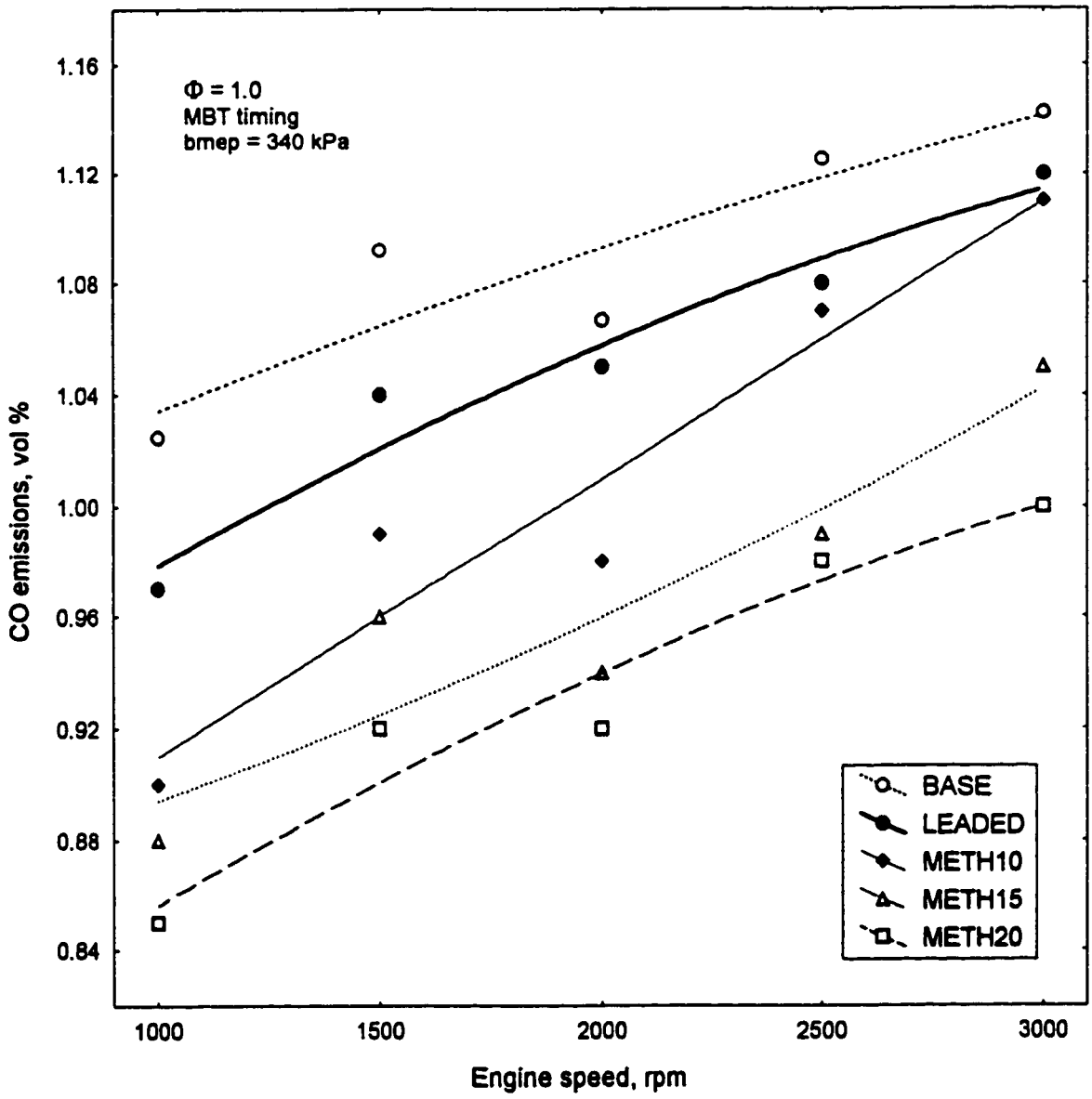

Figure 5-35: CO emissions vs. speed at a constant load of 80 Nm (bmep = 340 kPa) for the methanol blends.

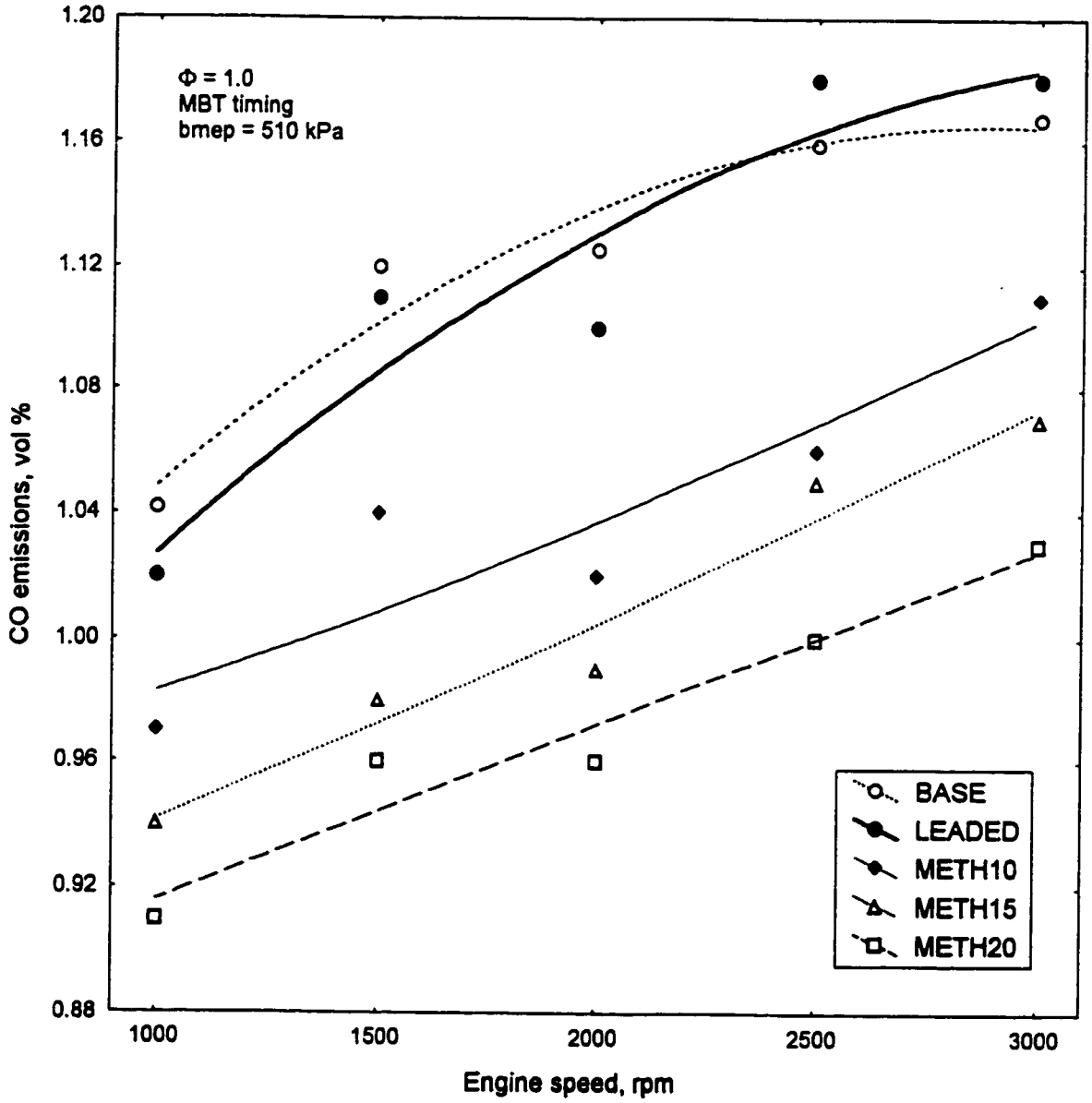

Figure 5-36: CO emissions vs. speed at a constant Load of 120 Nm (bmep = 510 kPa) for the methanol blends.

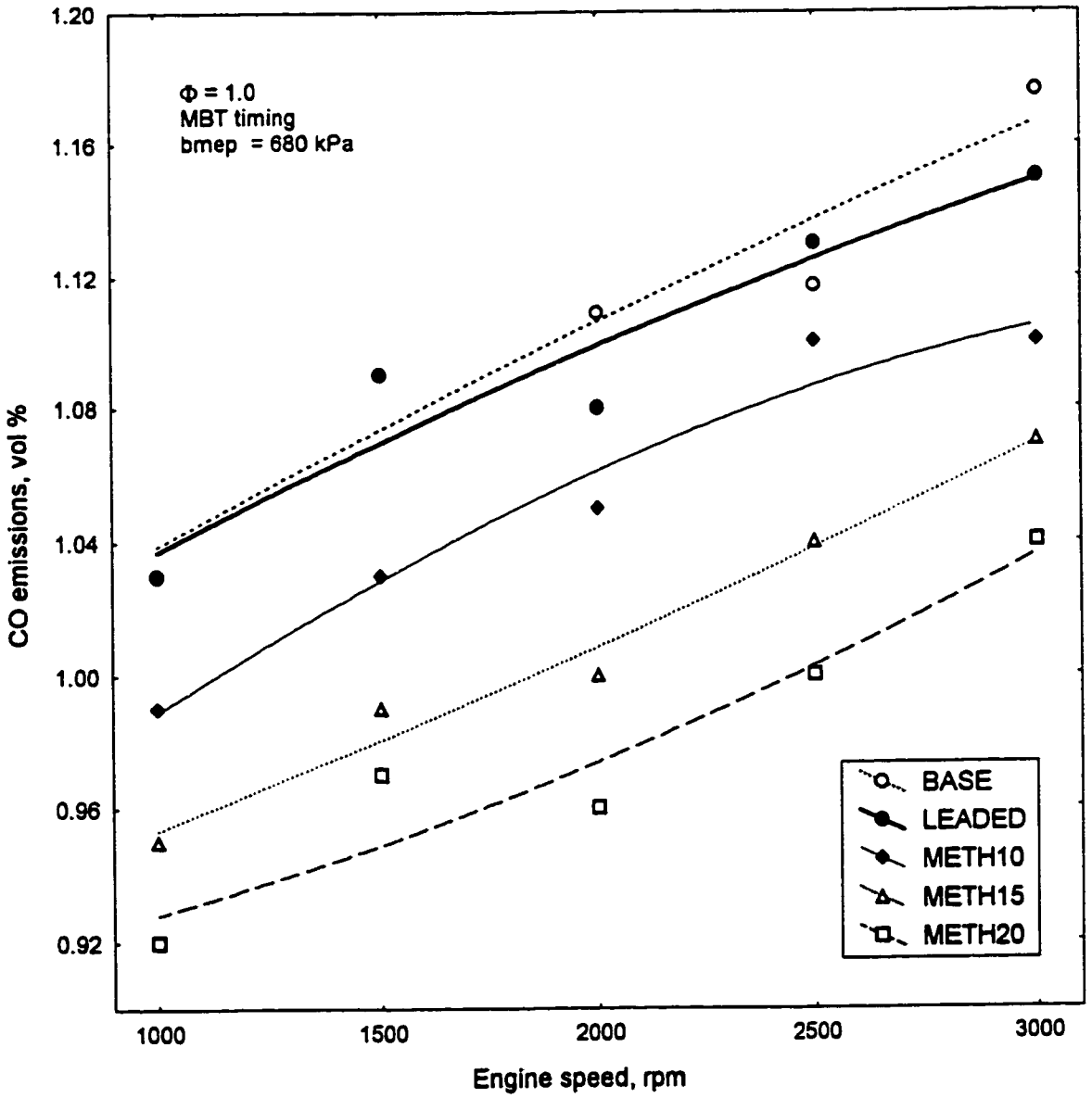

Figure 5-37: CO emissions vs. speed at a constant load of 160 Nm (bmep = 680 kPa) for the methanol blends.

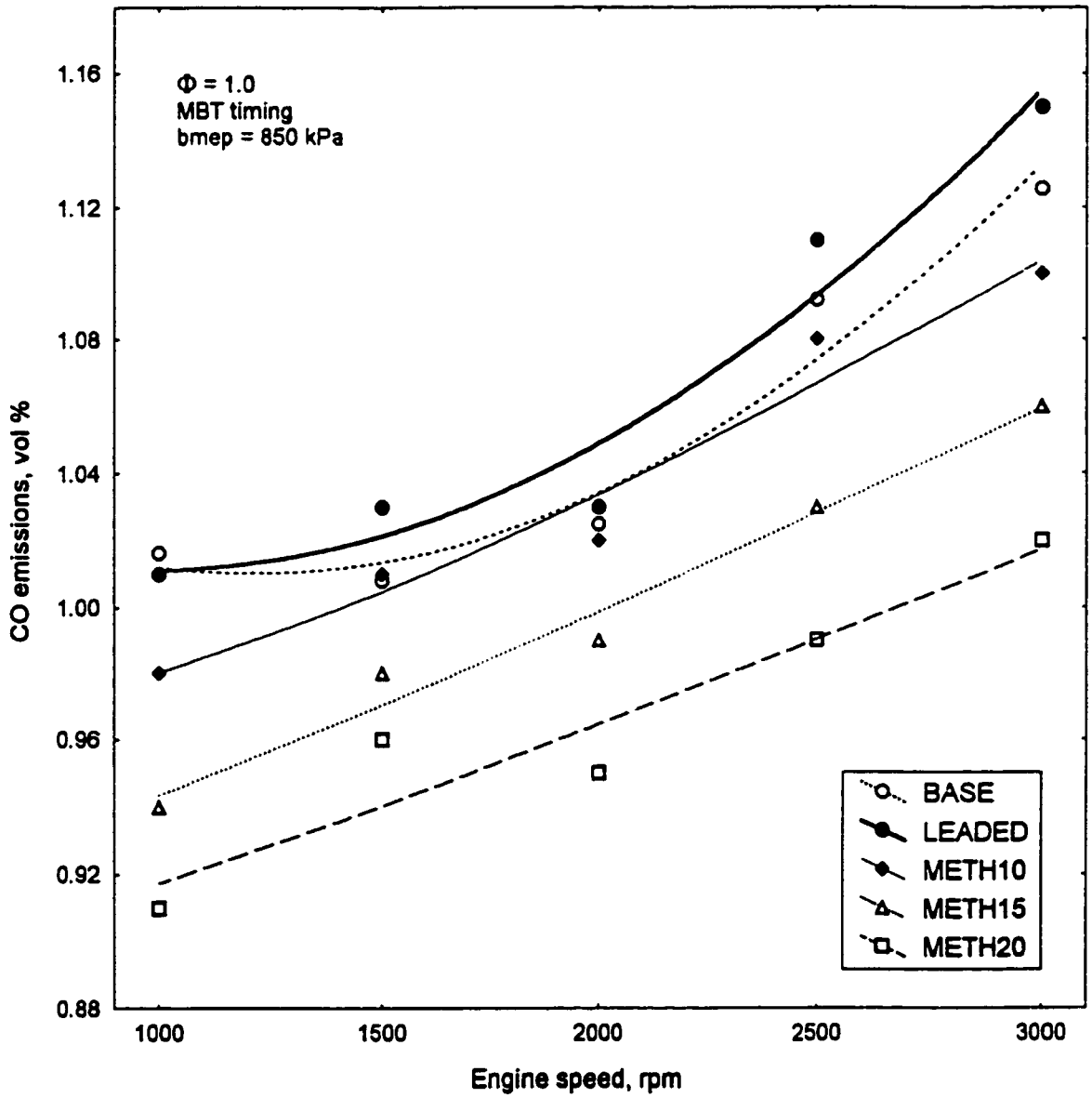

Figure 5-38: CO emissions vs. speed at a constant load of 200 Nm (bmep = 850 kPa) for the methanol blends.

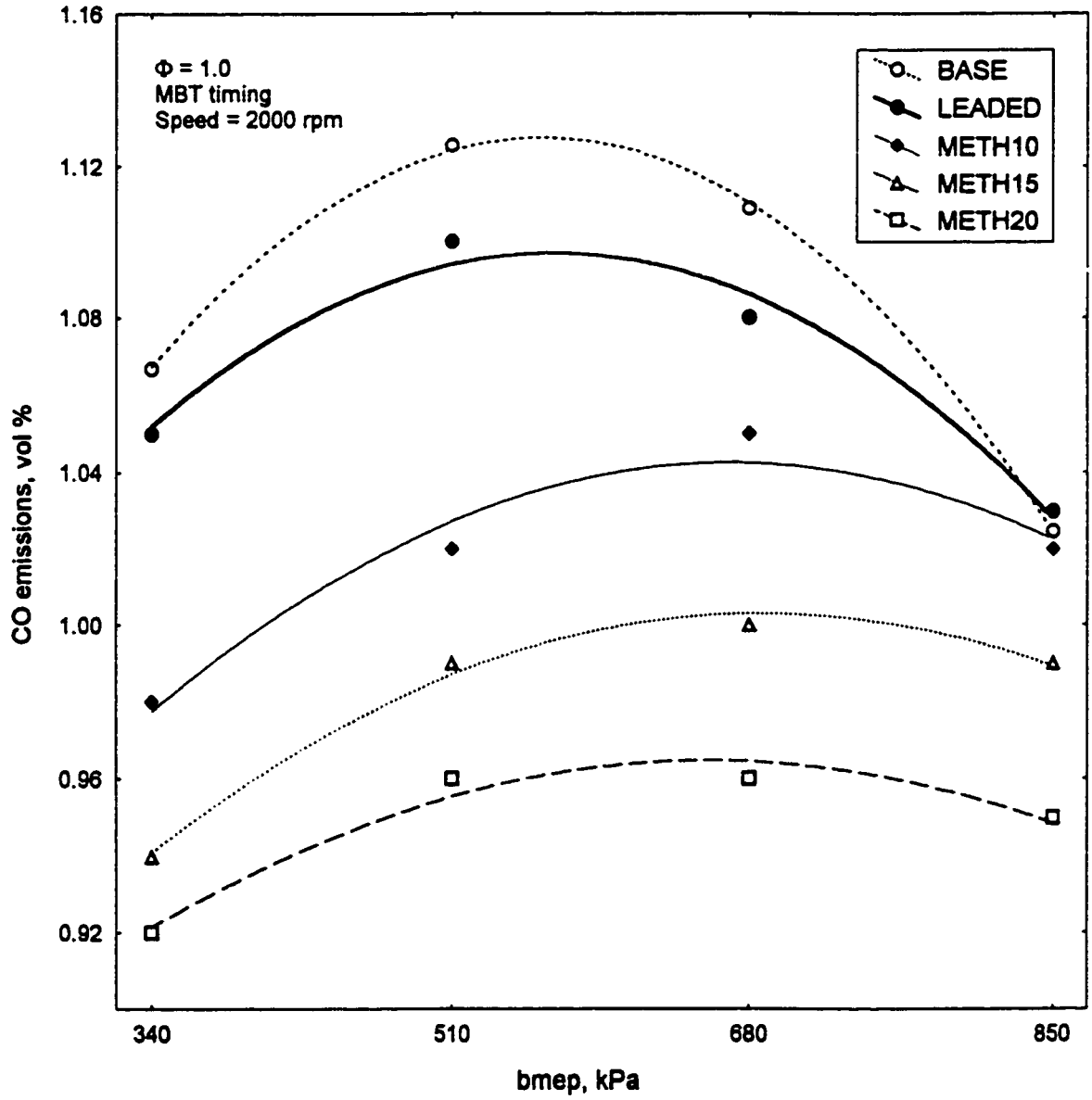

Figure 5-39: Effect of load on CO emissions at a constant speed of 2000 rpm for the methanol blends.

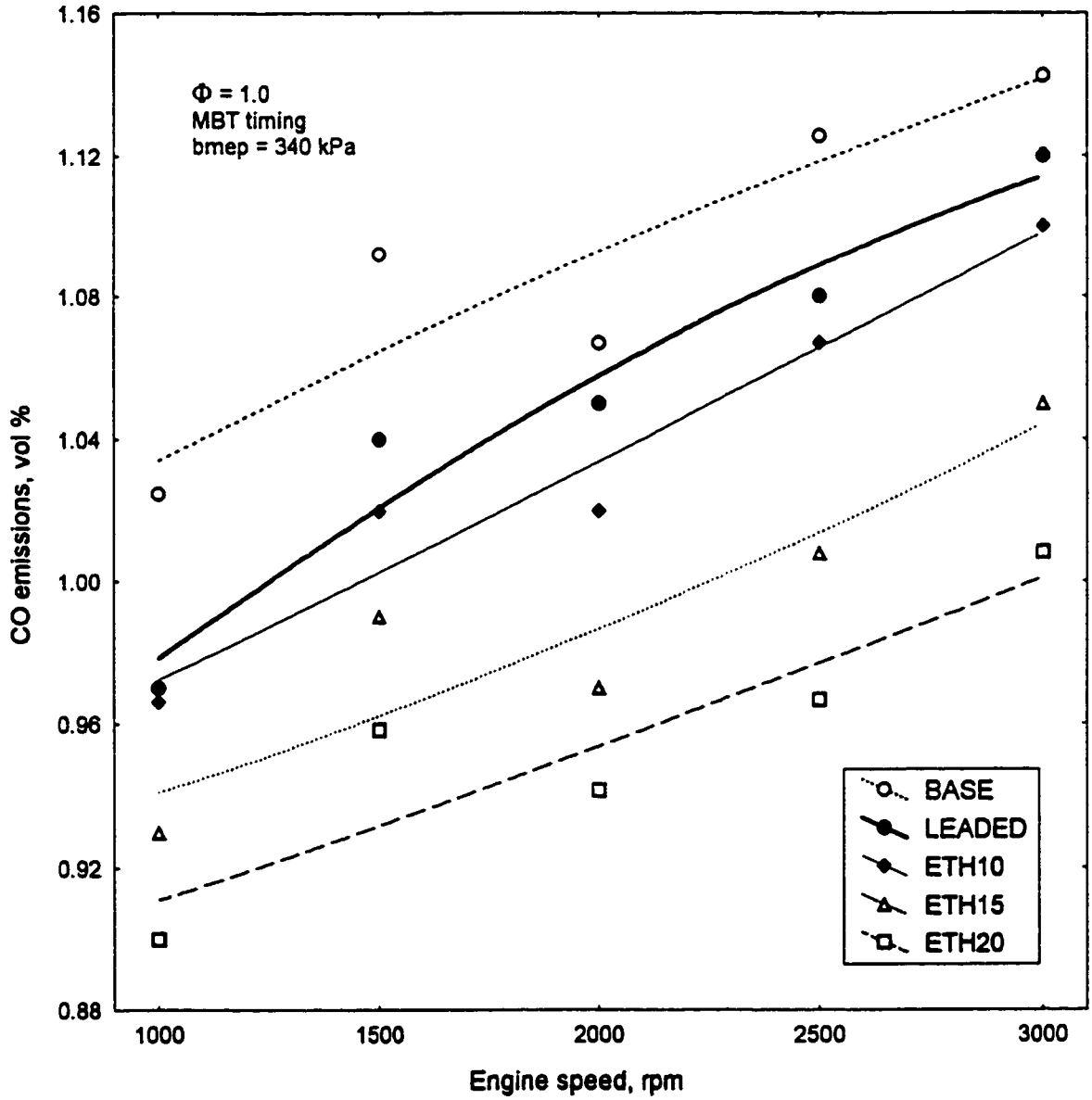

Figure 5-40: CO emissions vs. speed at a constant load of 80 Nm (bmep = 340 kPa) for the ethanol blends.

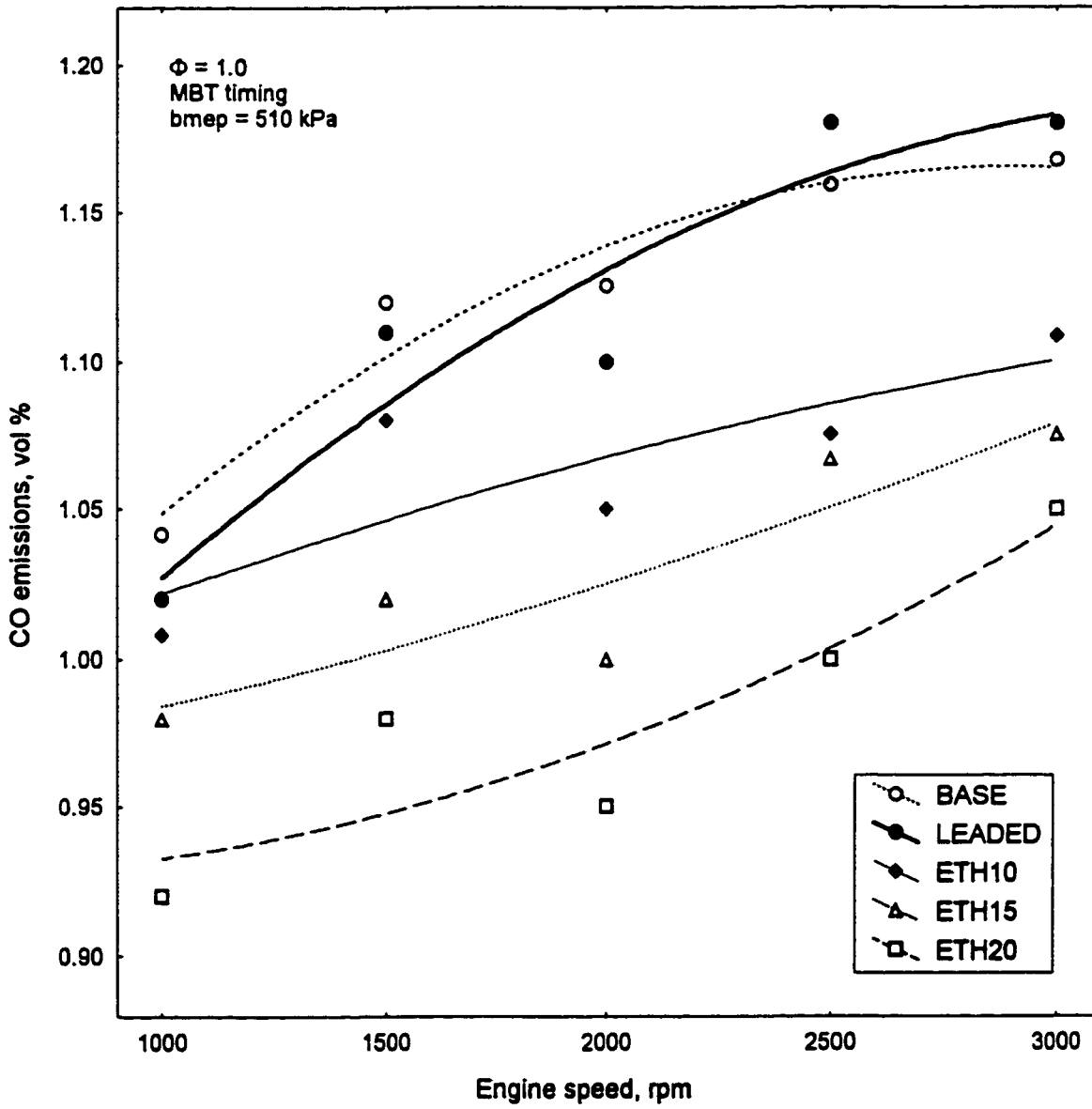

Figure 5-41: CO emissions vs. speed at a constant Load of 120 Nm (bmep = 510 kPa) for the ethanol blends.

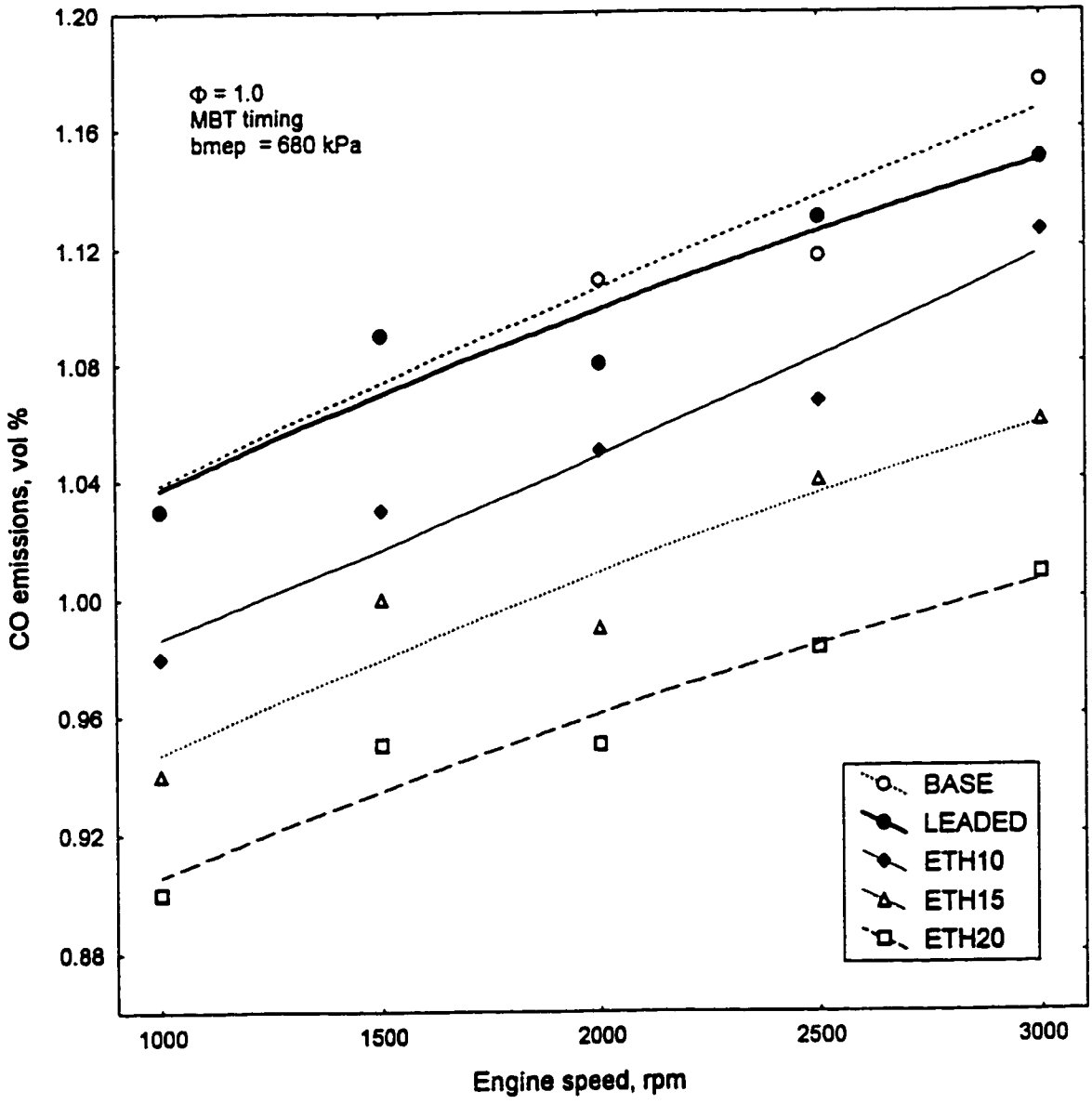

Figure 5-42: CO emissions vs. speed at a constant load of 160 Nm (bmep = 680 kPa) for the ethanol blends.

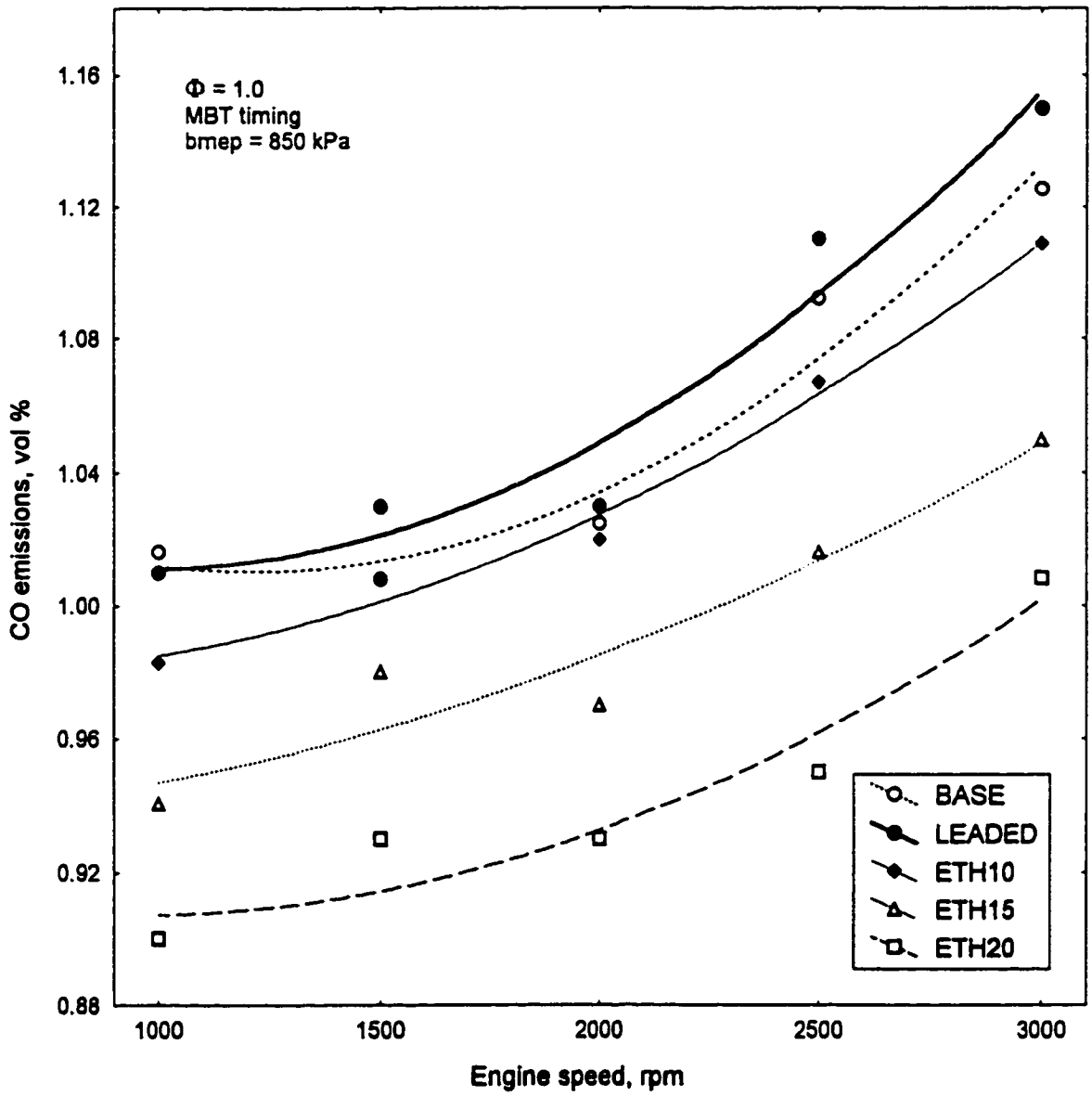

Figure 5-43: CO emissions vs. speed at a constant load of 200 Nm (bmep = 850 kPa) for the ethanol blends.

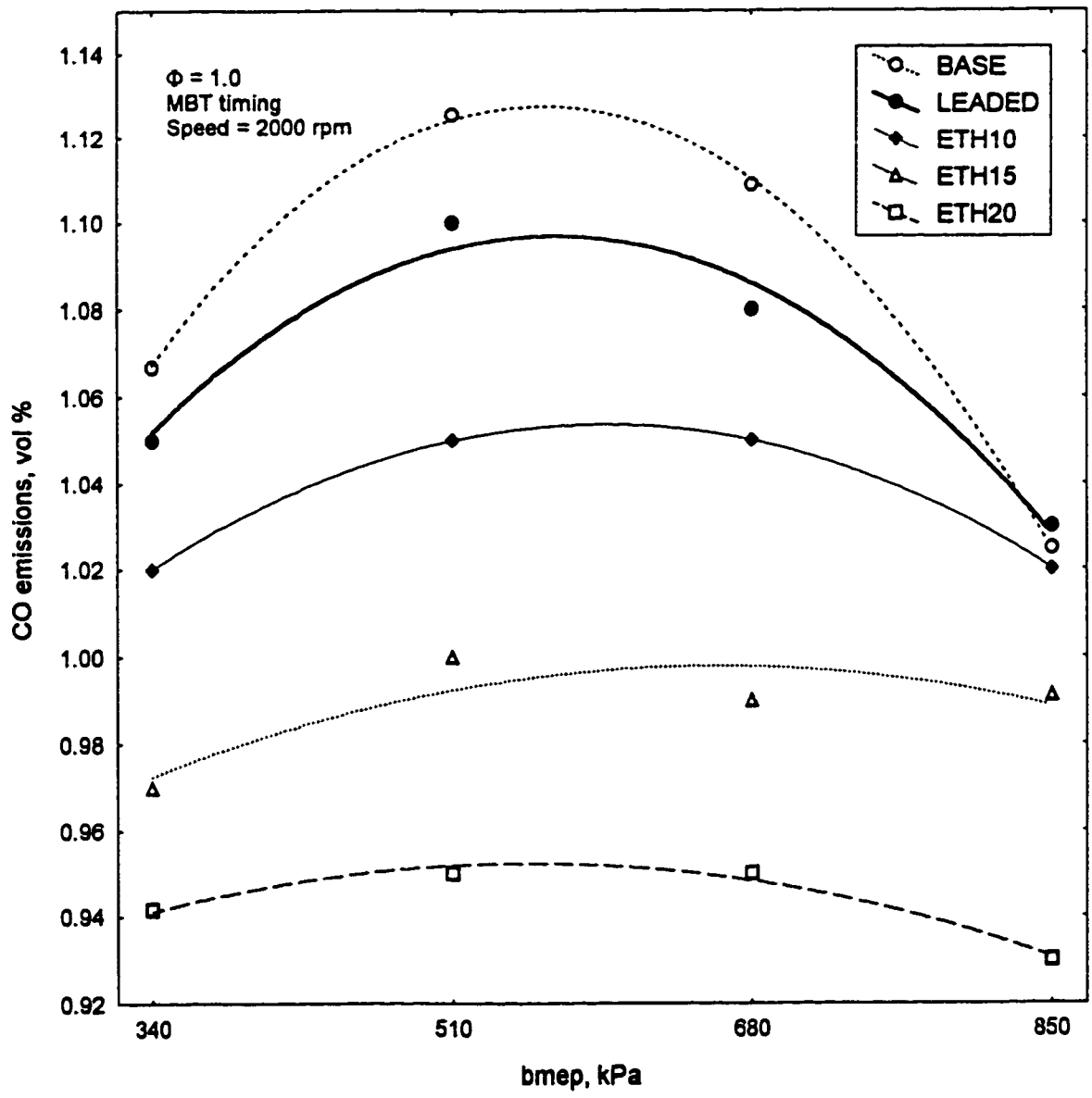

Figure 5-44: Effect of load on CO emissions at a constant speed of 2000 rpm for the ethanol blends.

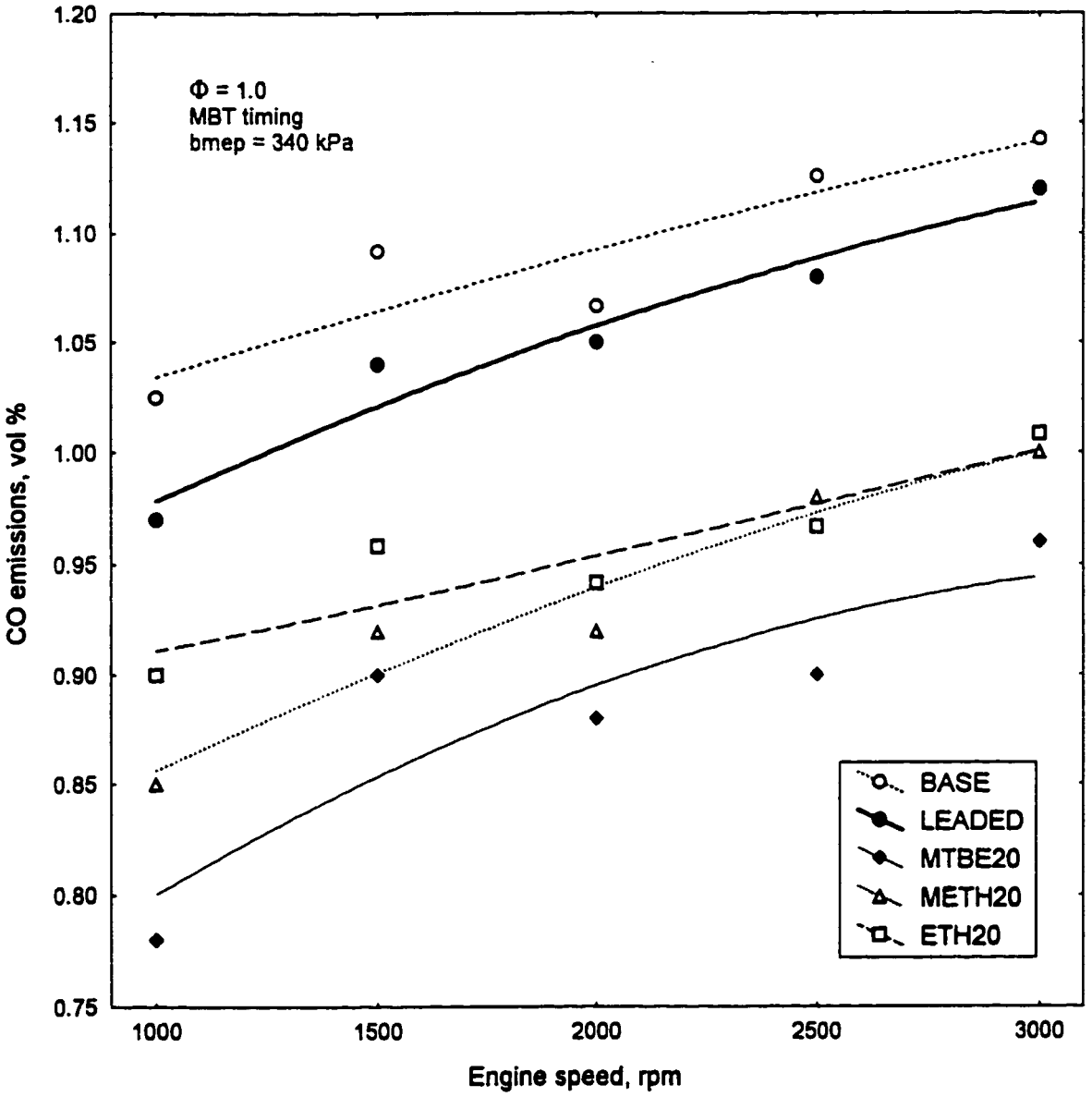

Figure 5-45: CO emissions vs. speed at a constant load of 80 Nm (bmep = 340 kPa) for the least emitting blends compared to the base and leaded fuels.

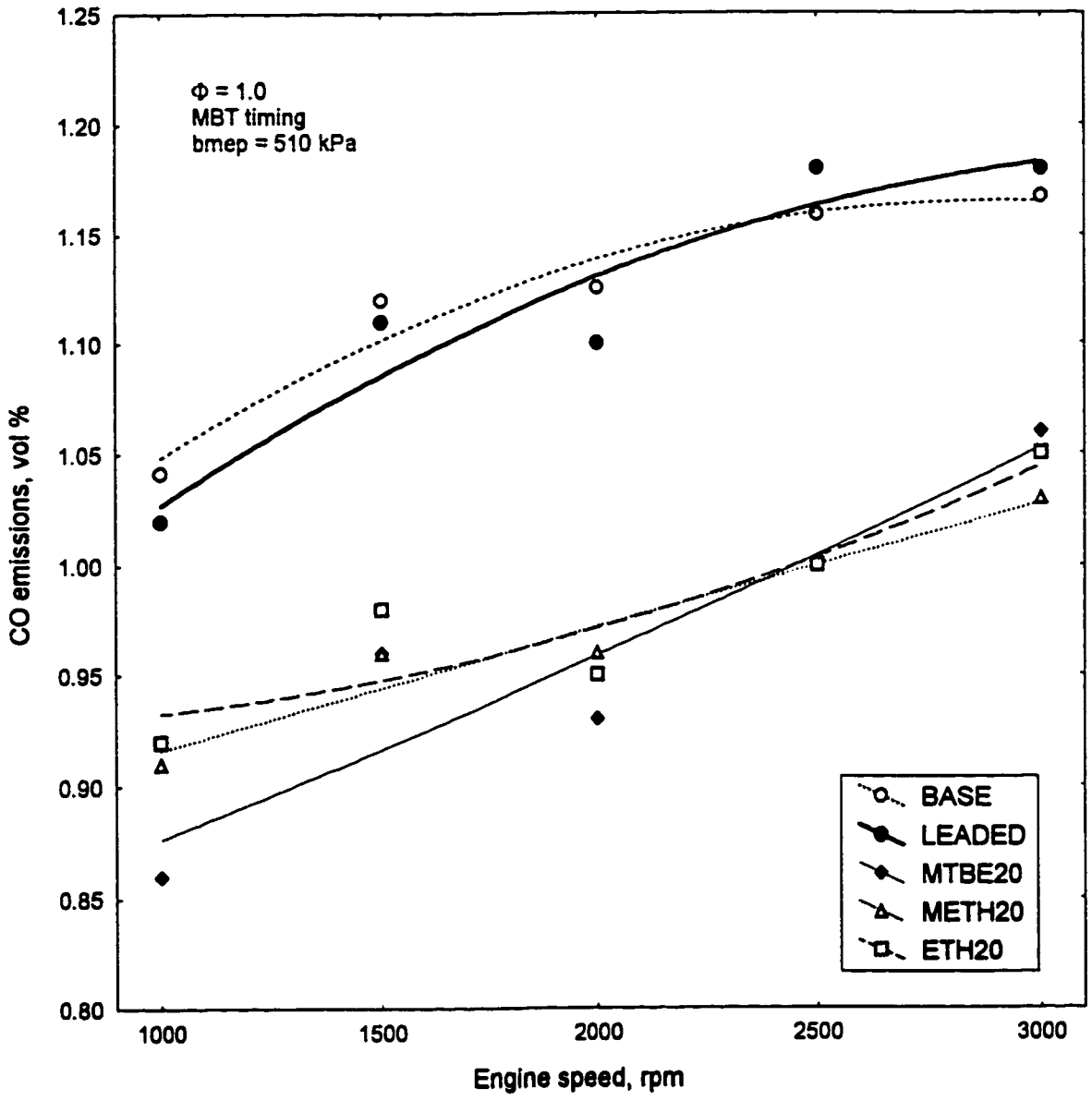

Figure 5-46: CO emissions vs. speed at a constant Load of 120 Nm (bmep = 510 kPa) for the least emitting blends compared to the base and leaded fuels.

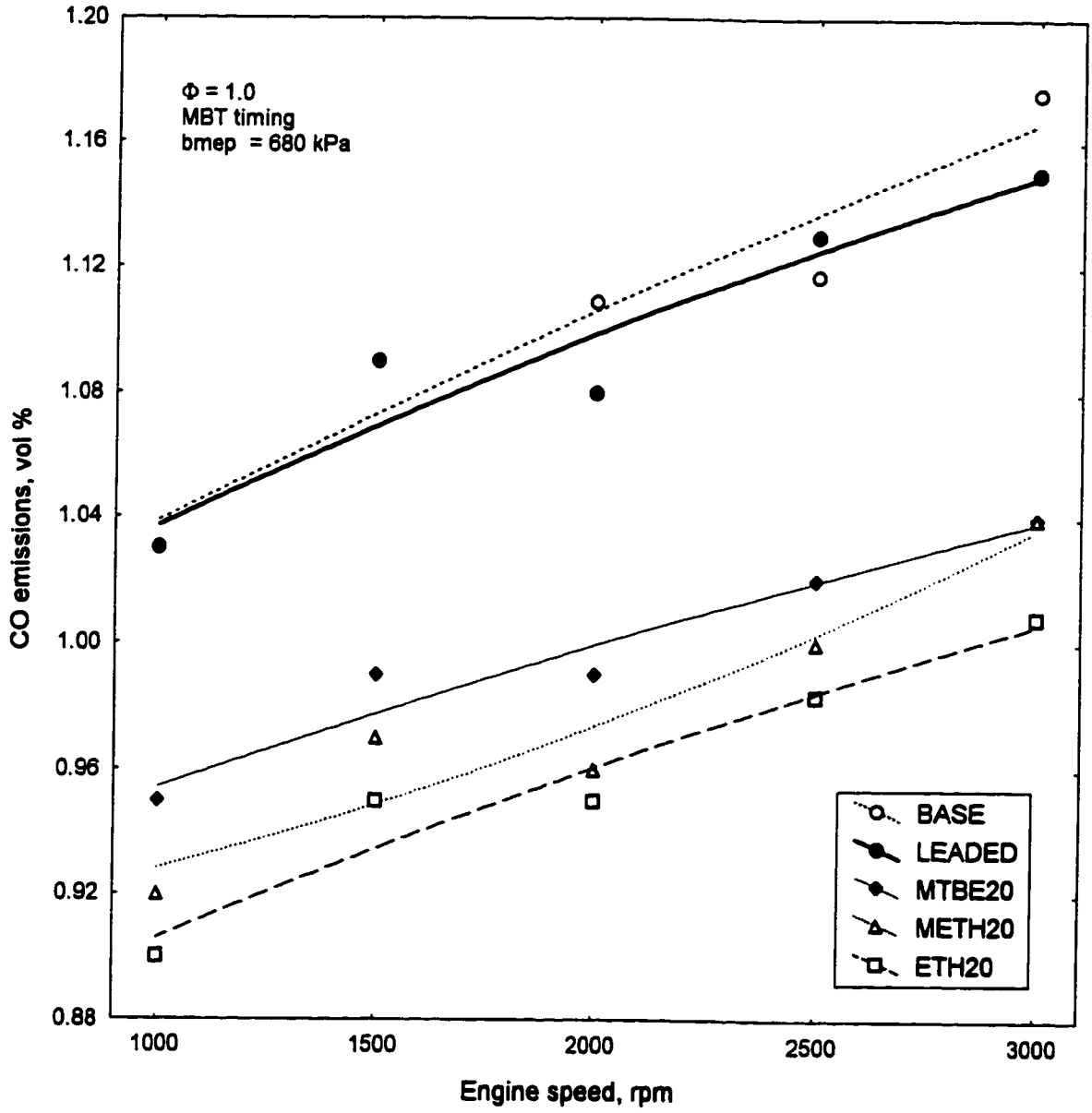

Figure 5-47: CO emissions vs. speed at a constant load of 160 Nm (bmep = 680 kPa) for the least emitting blends compared to the base and leaded fuels.

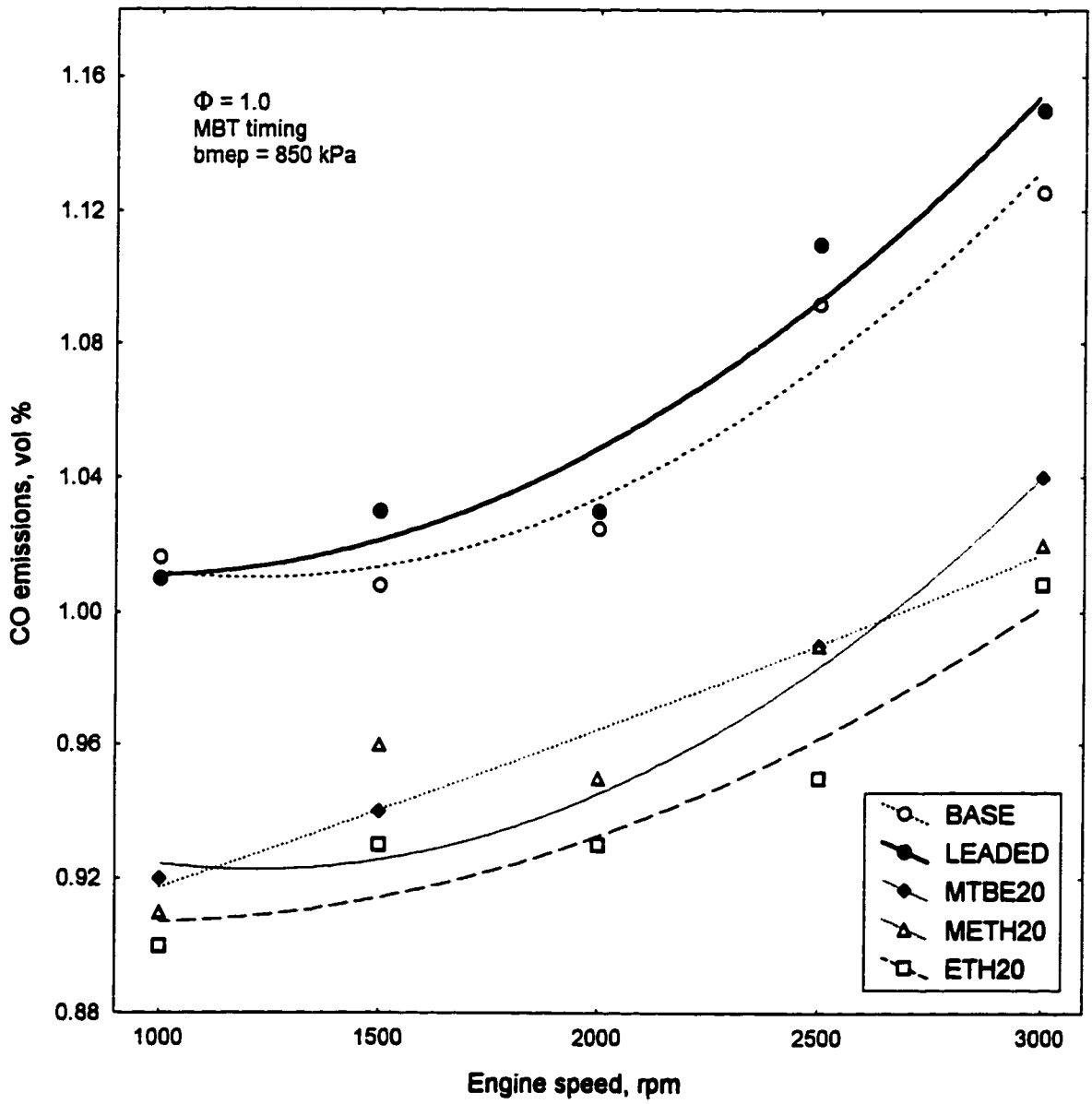

Figure 5-48: CO emissions vs. speed at a constant load of 200 Nm (bmep = 850 kPa) for the least emitting blends compared to base and leaded fuels.

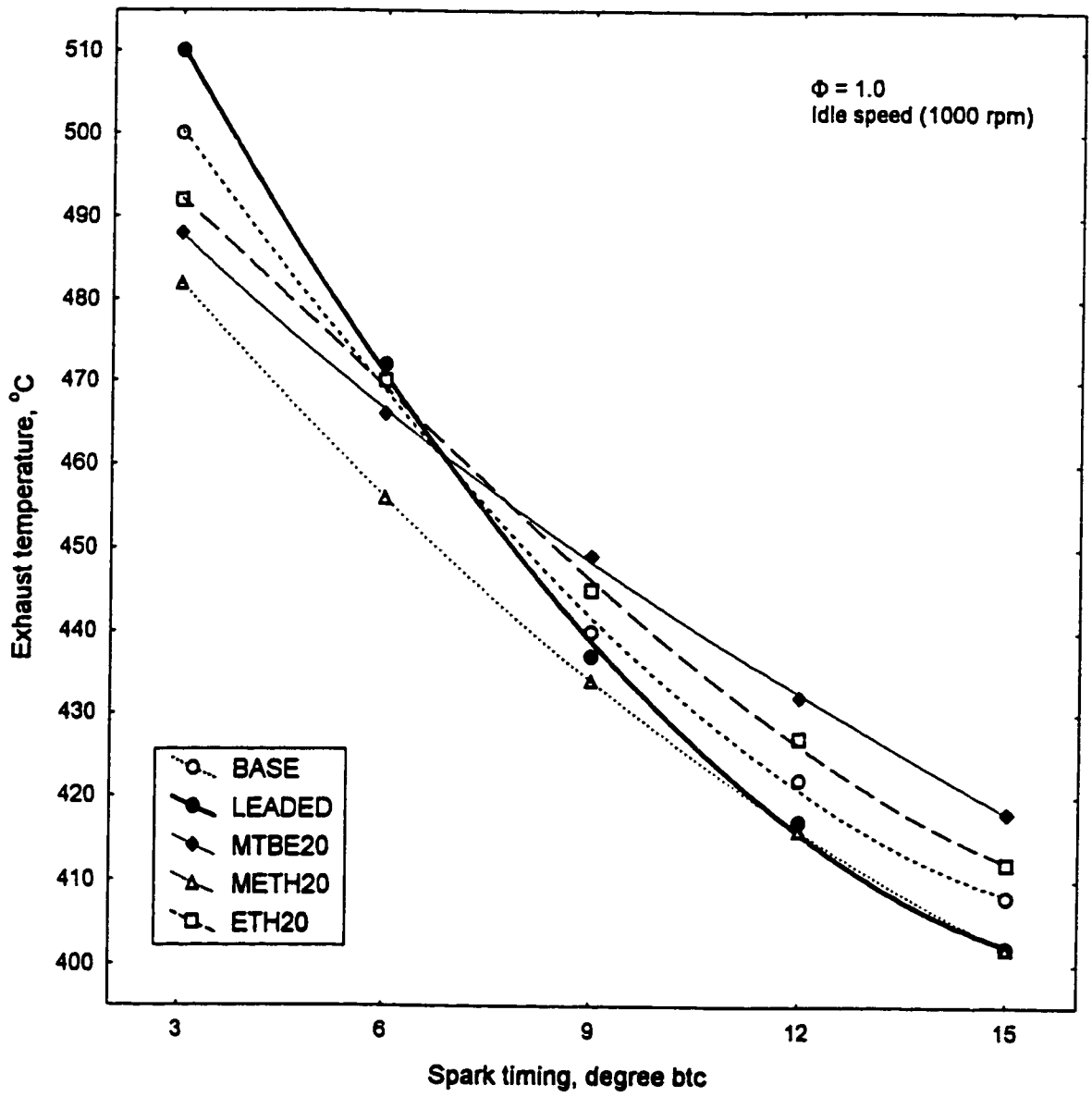

Figure 5-49: Exhaust gas temperatures at idle speed as a function of spark timing for selected blends.

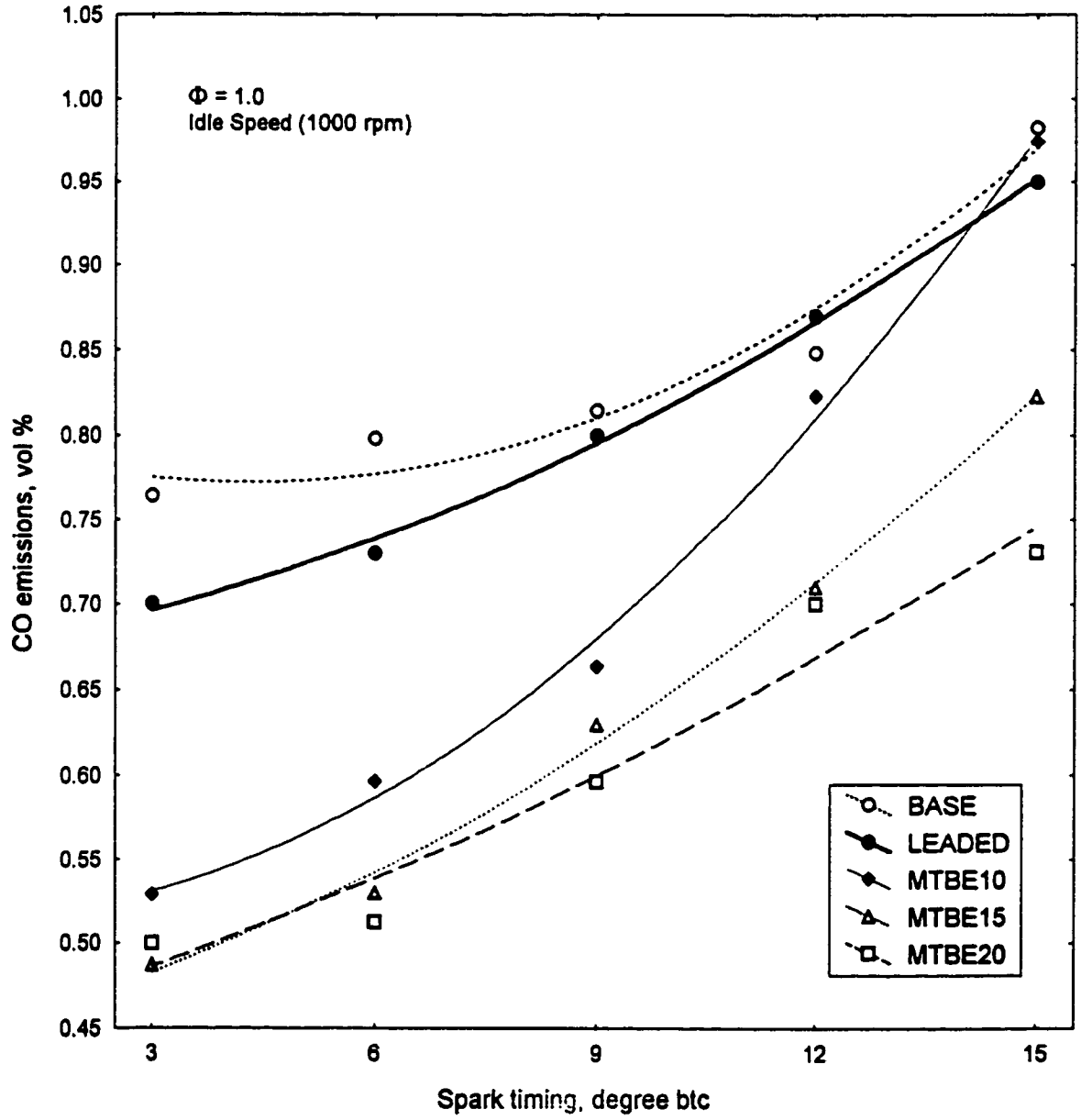

Figure 5-50: Effect of spark timing on CO emissions at idle speed for the MTBE blends.

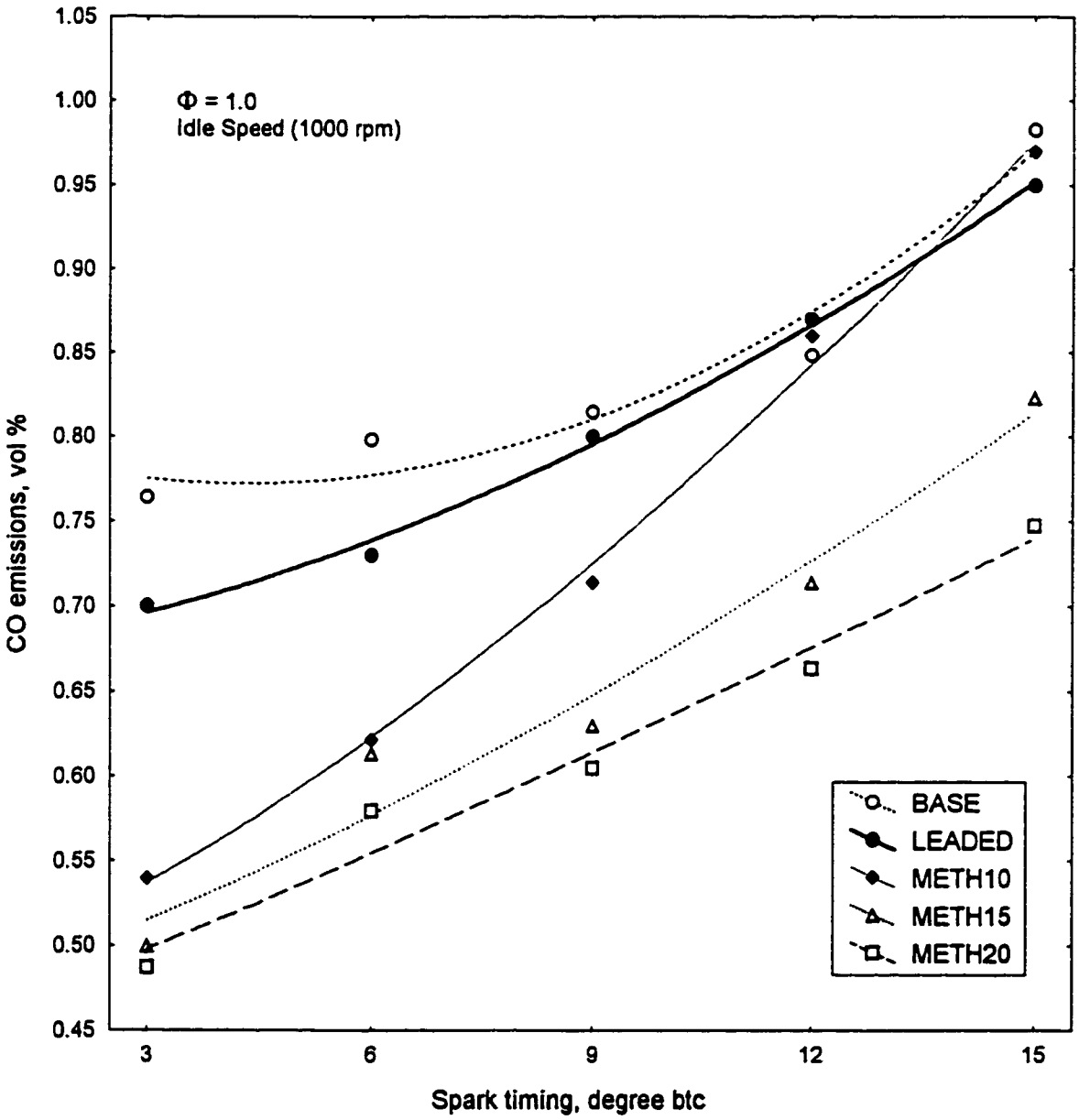

Figure 5-51: Effect of spark timing on CO emissions at idle speed for the methanol blends.

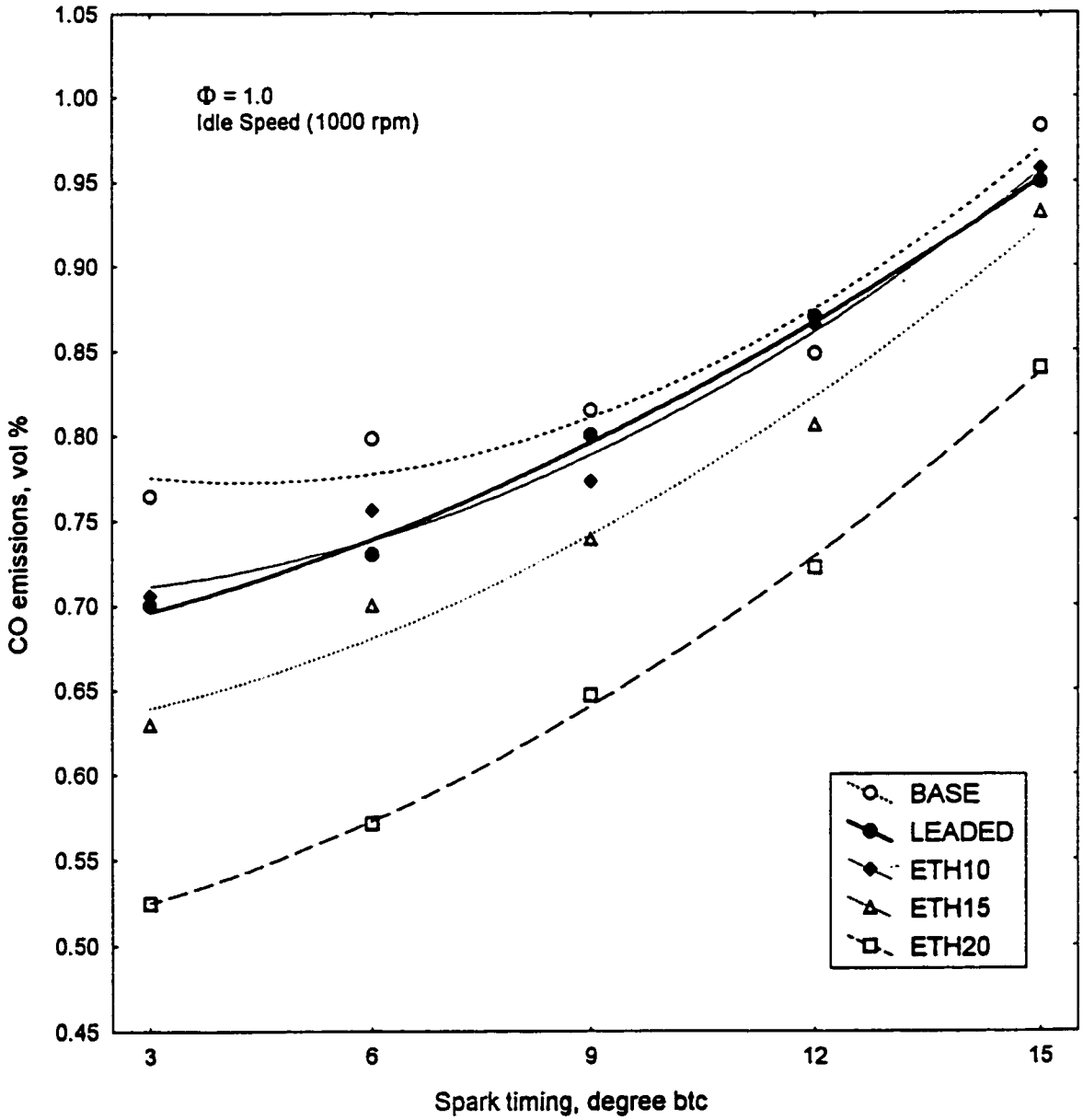

Figure 5-52: Effect of spark timing on CO emissions at idle speed for the ethanol blends.

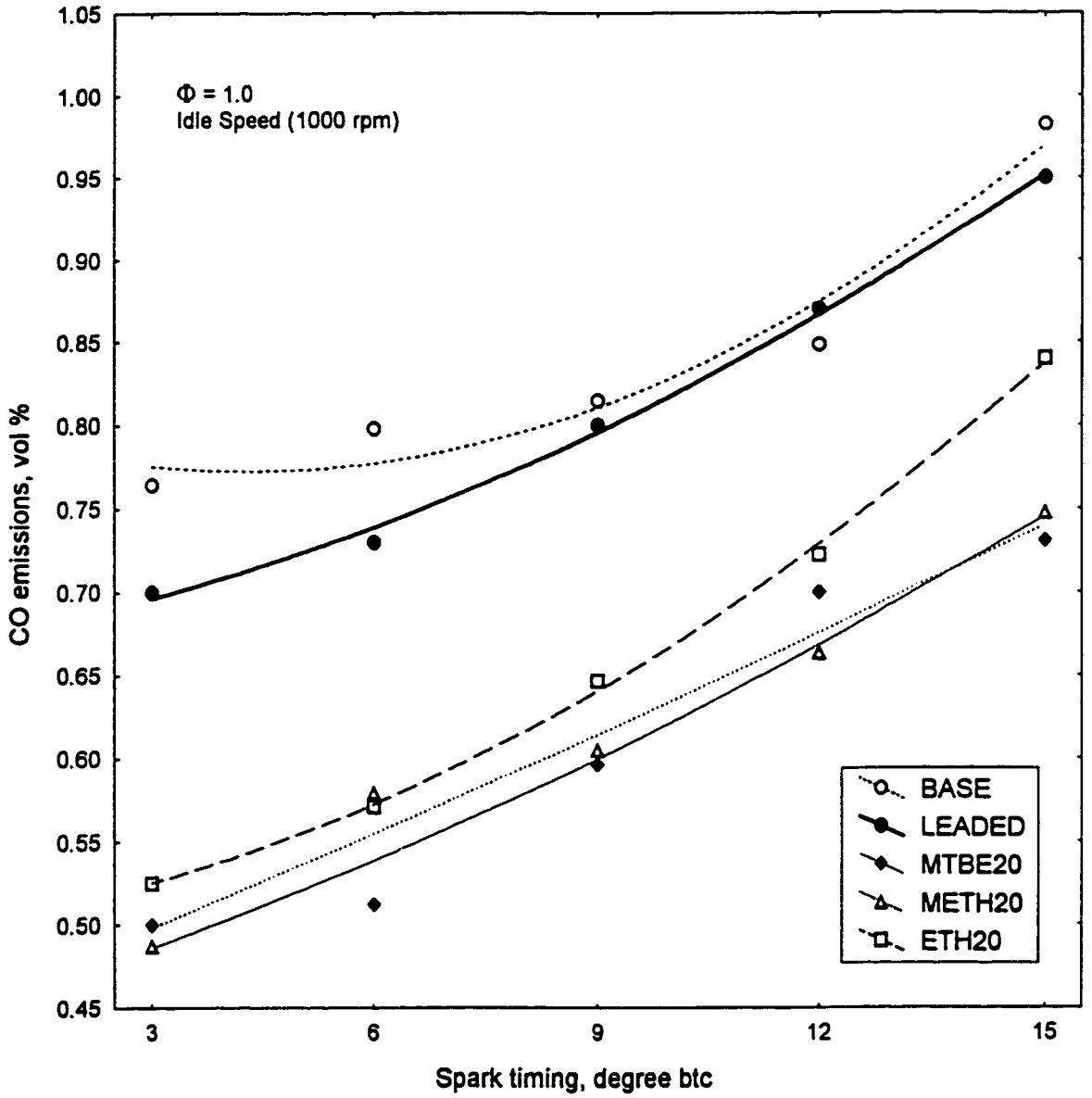

Figure 5-53: Effect of spark timing on CO emissions at idle speed for the least emitting blends compared to the base and leaded fuels.

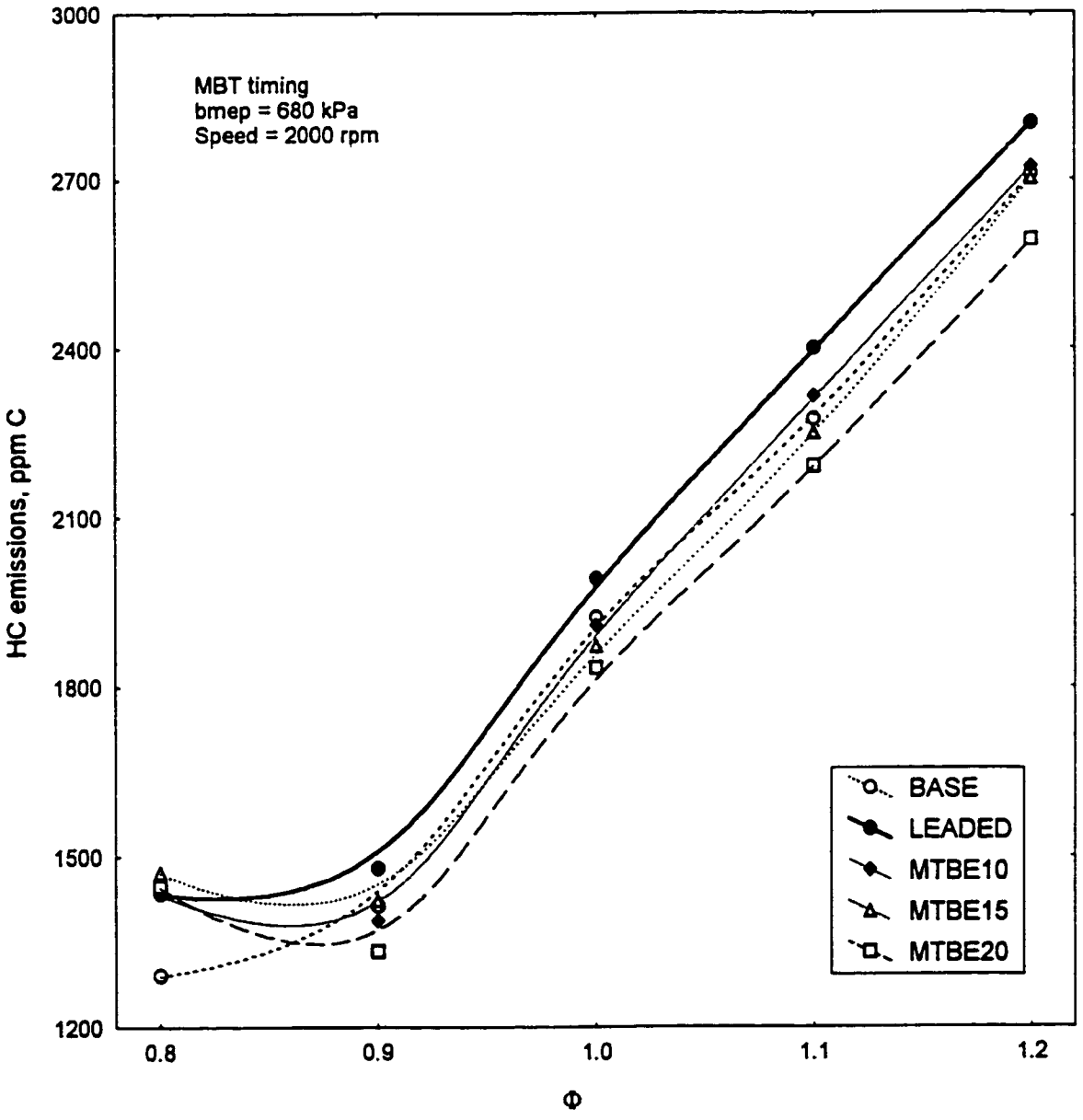

Figure 5-54: Effect of equivalence ratio on HC emissions for the MTBE blends.

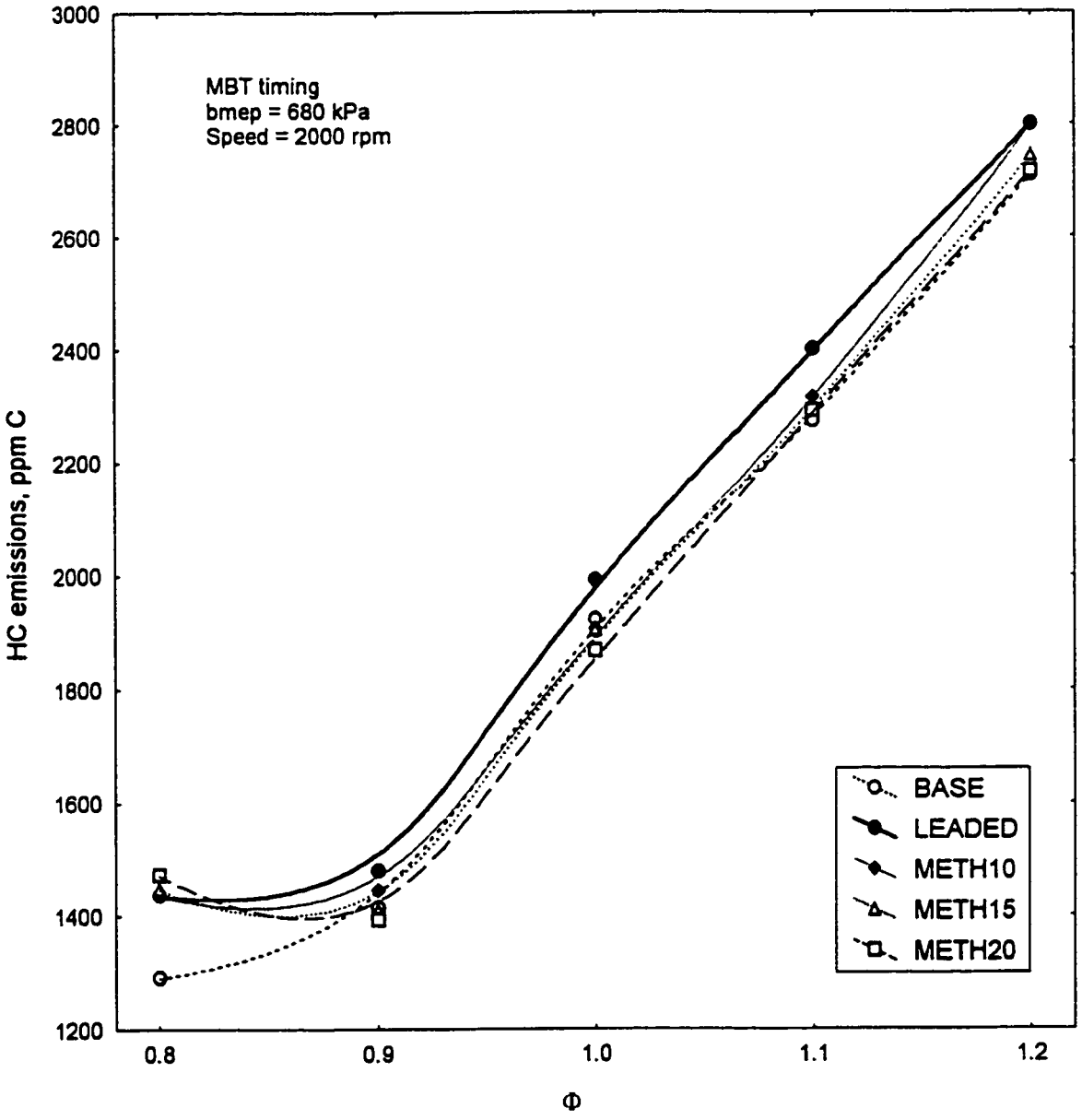

Figure 5-55: Effect of equivalence ratio on HC emissions for the methanol blends.

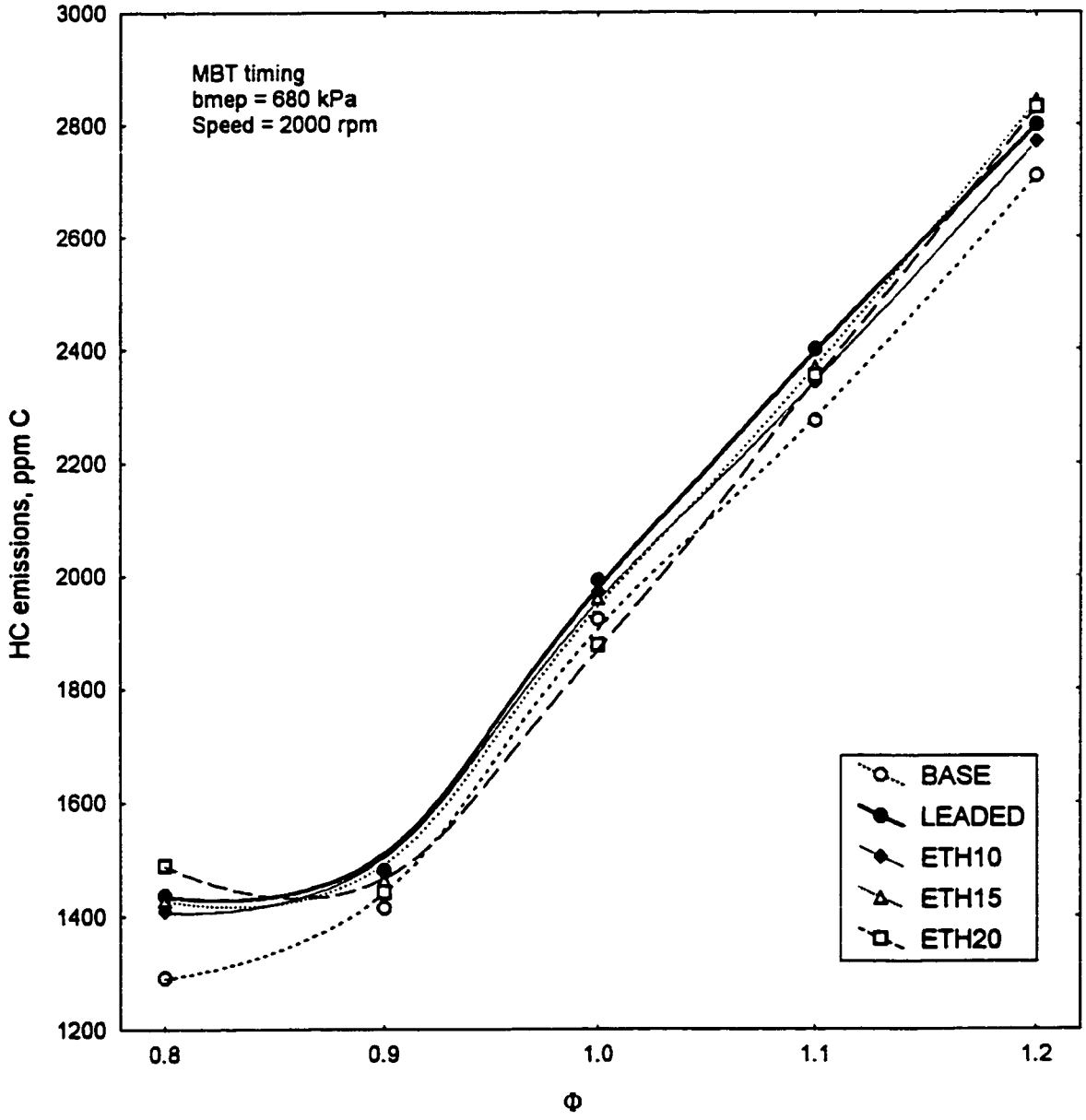

Figure 5-56: Effect of equivalence ratio on HC emissions for the ethanol blends.

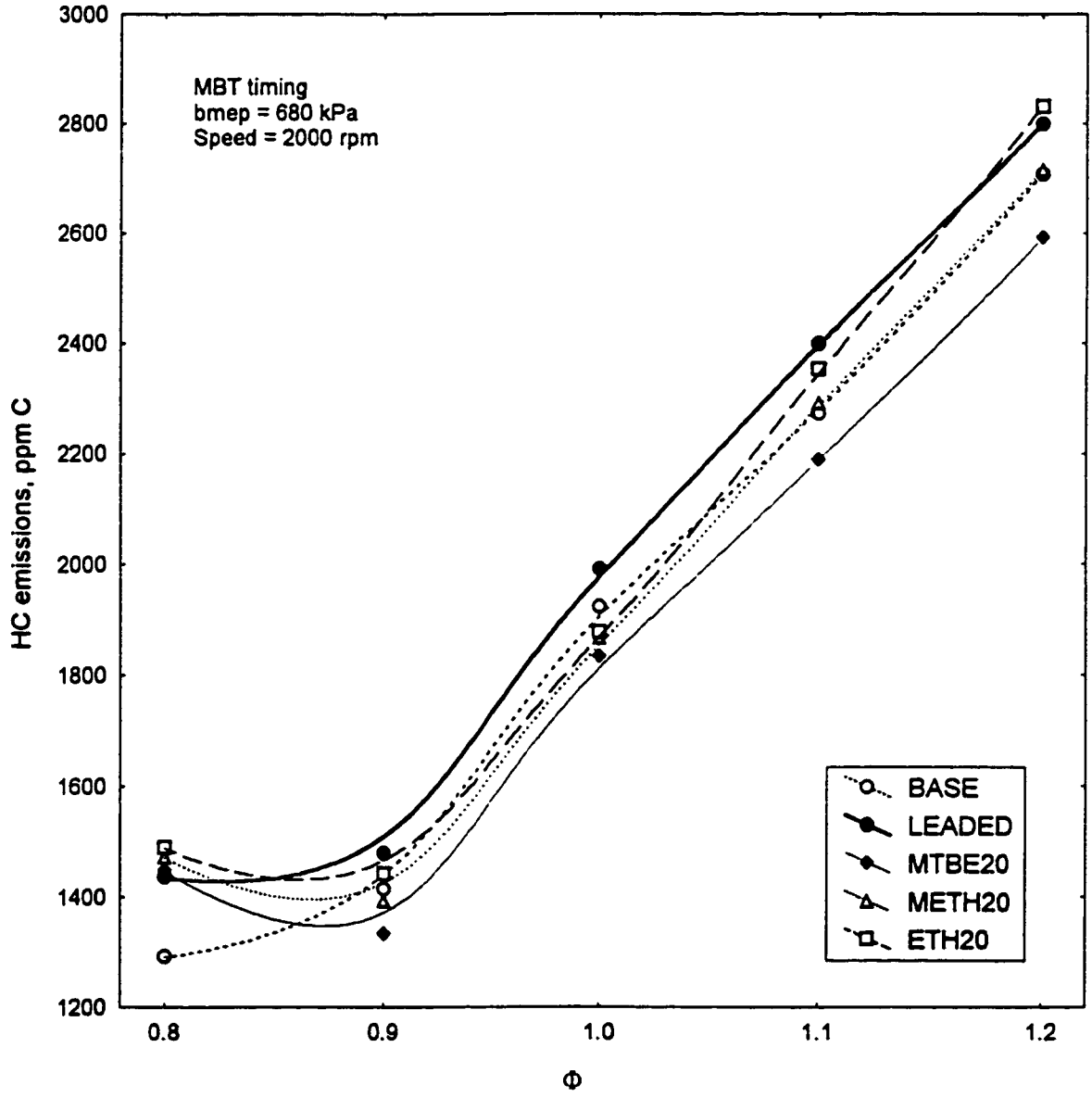

Figure 5-57: Effect of equivalence ratio on HC emissions for the least emitting blends compared to the base and leaded fuels.

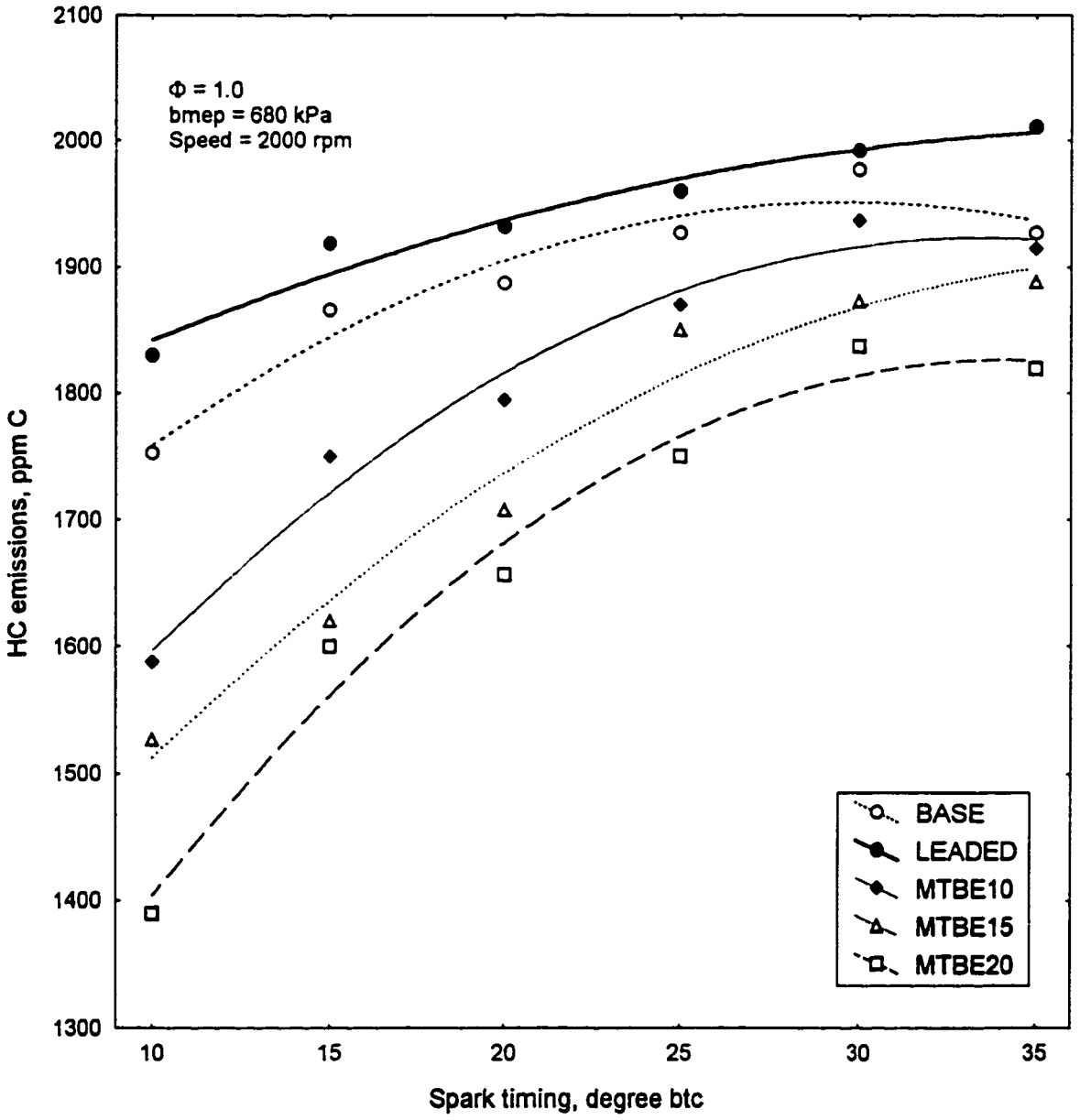

Figure 5-58: Effect of spark timing on HC emissions for the MTBE blends.

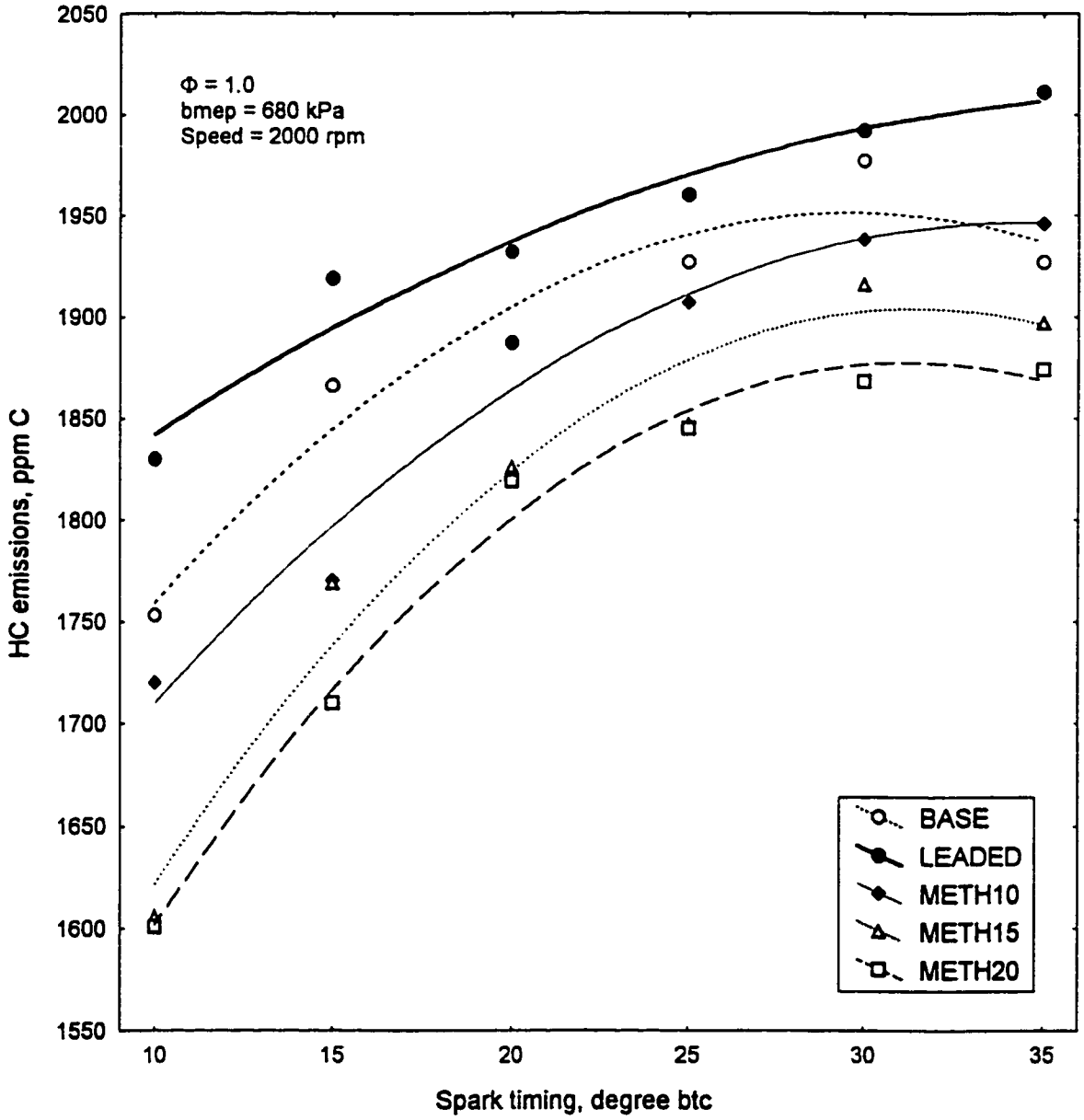

Figure 5-59: Effect of spark timing on HC emissions for the methanol blends.

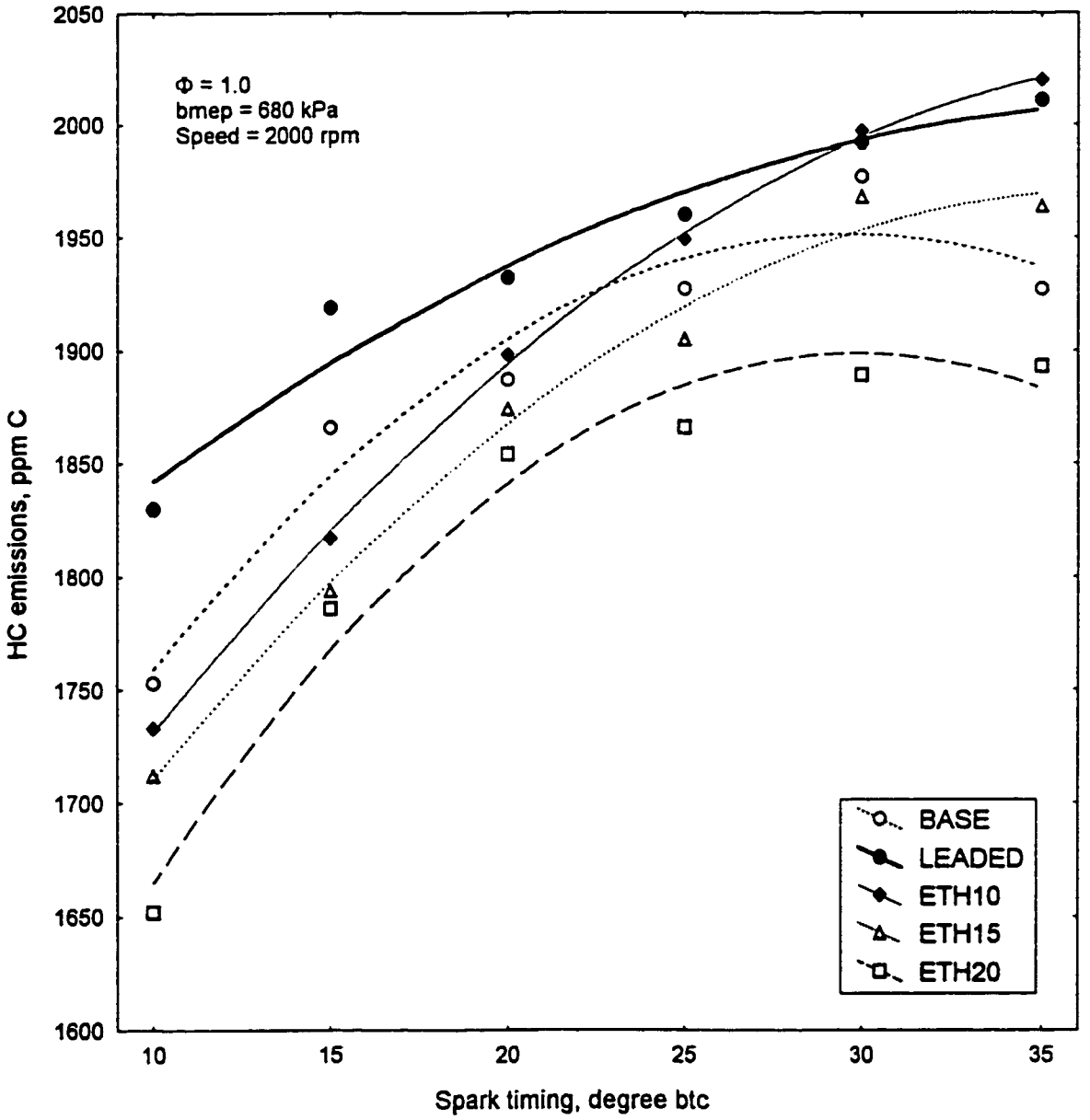

Figure 5-60: Effect of spark timing on HC emissions for the ethanol blends.

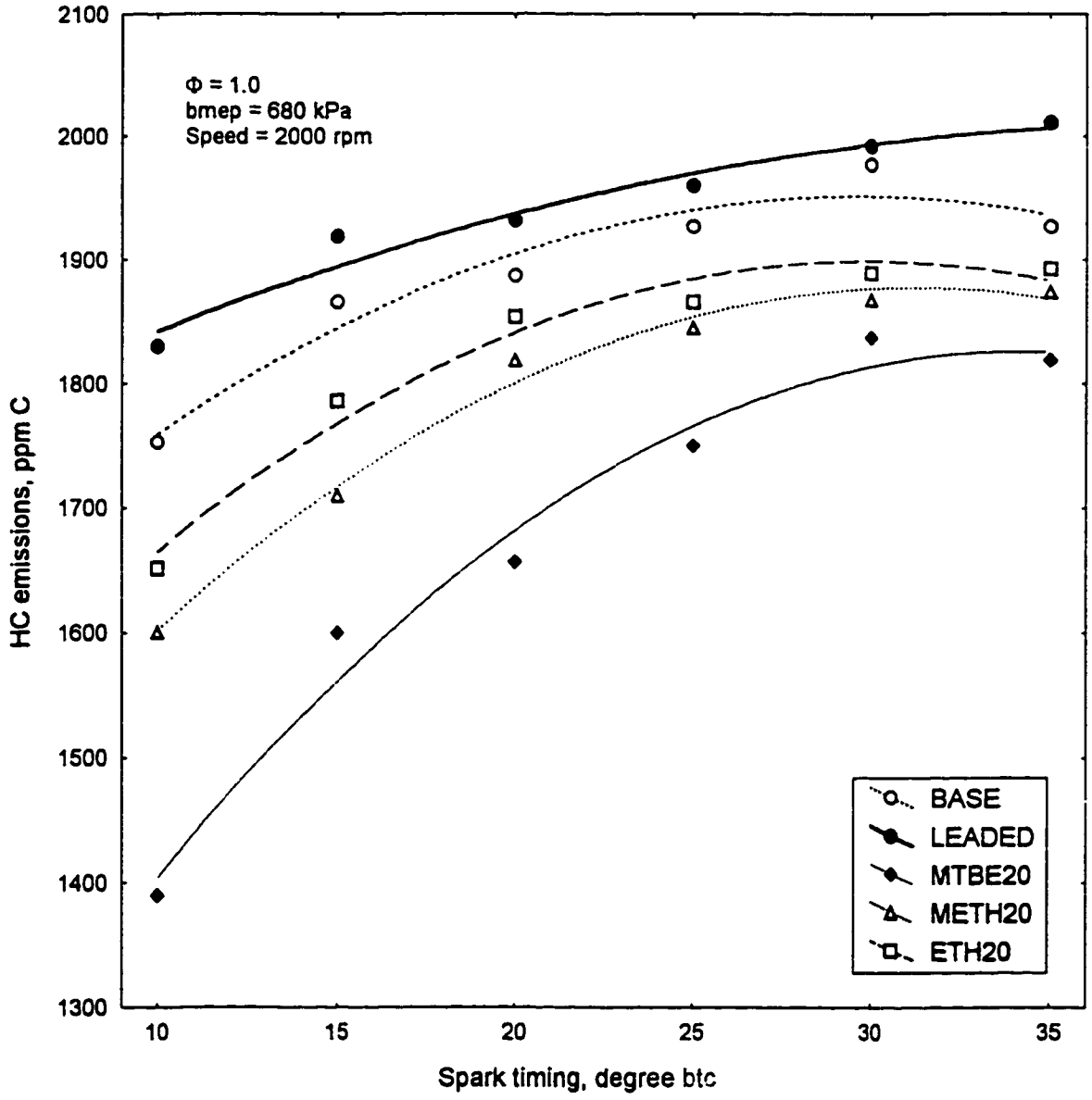

Figure 5-61: Effect of spark timing on HC emissions for the least emitting blends compared to the base and leaded fuels.

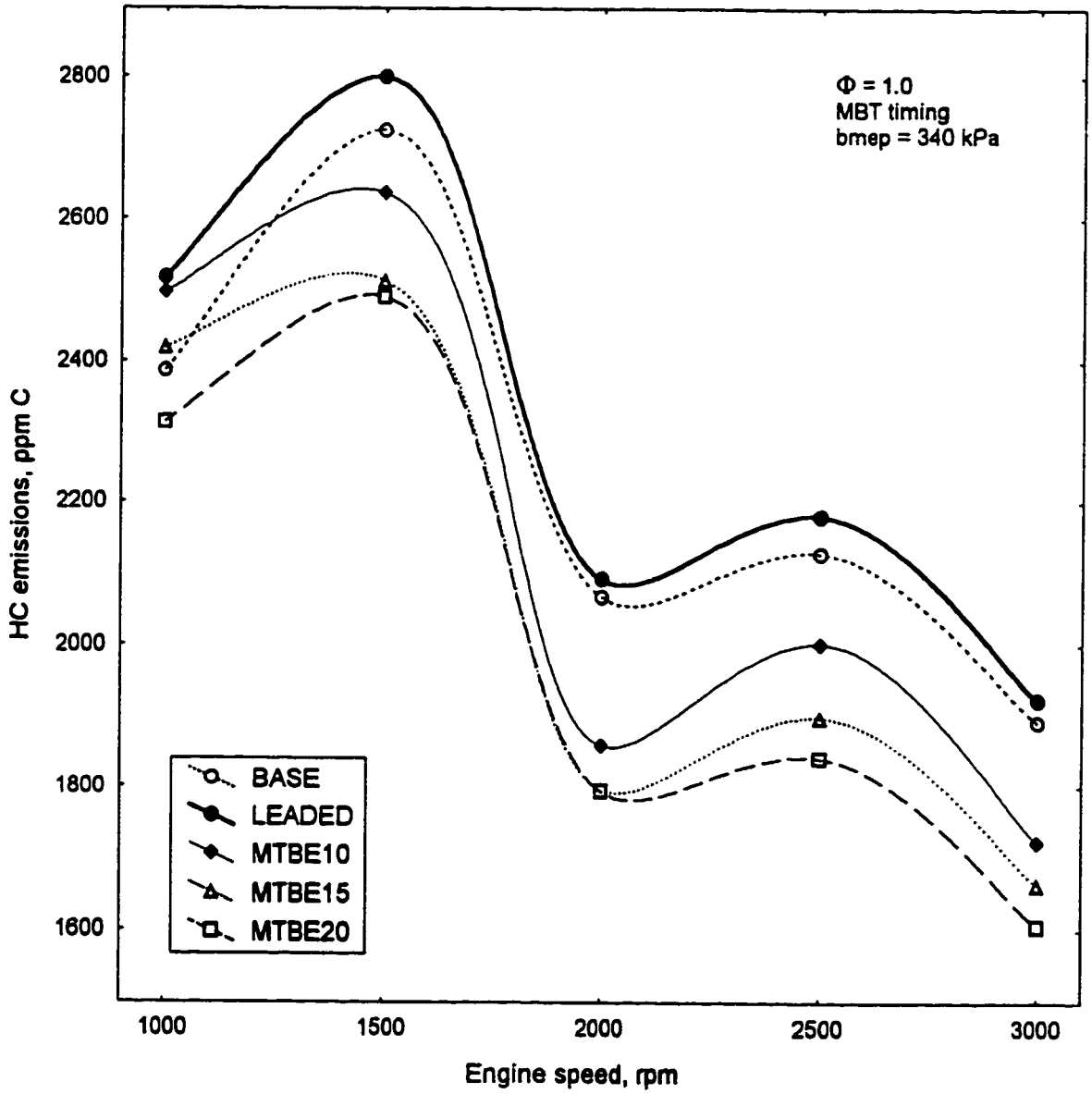

Figure 5-62: Exhaust HC emissions vs. speed at a constant load of 80 Nm (bmep = 340 kPa) for the MTBE blends.

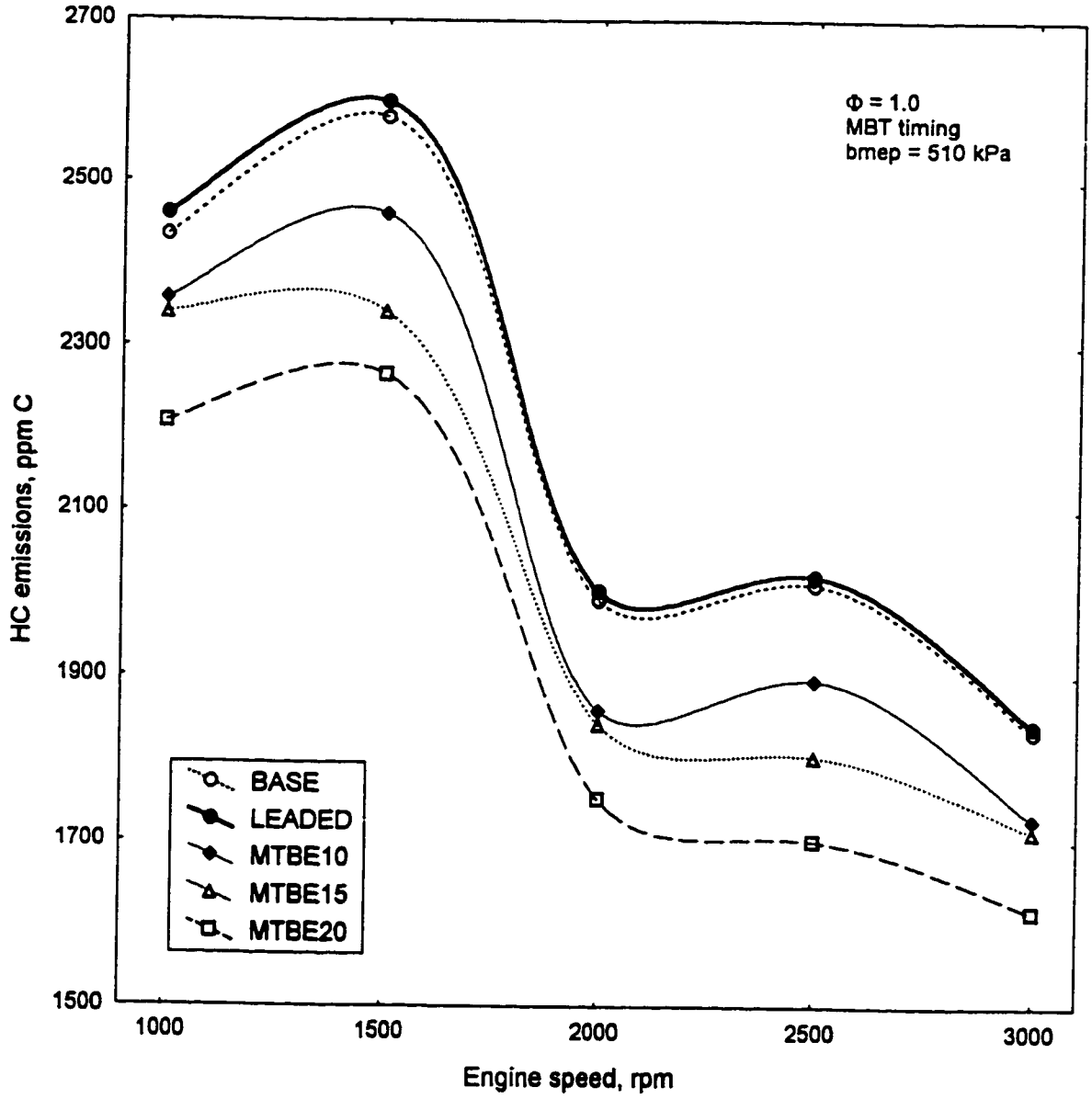

Figure 5-63: Exhaust HC emissions vs. speed at a constant load of 120 Nm (bmep = 510 kPa) for the MTBE blends.

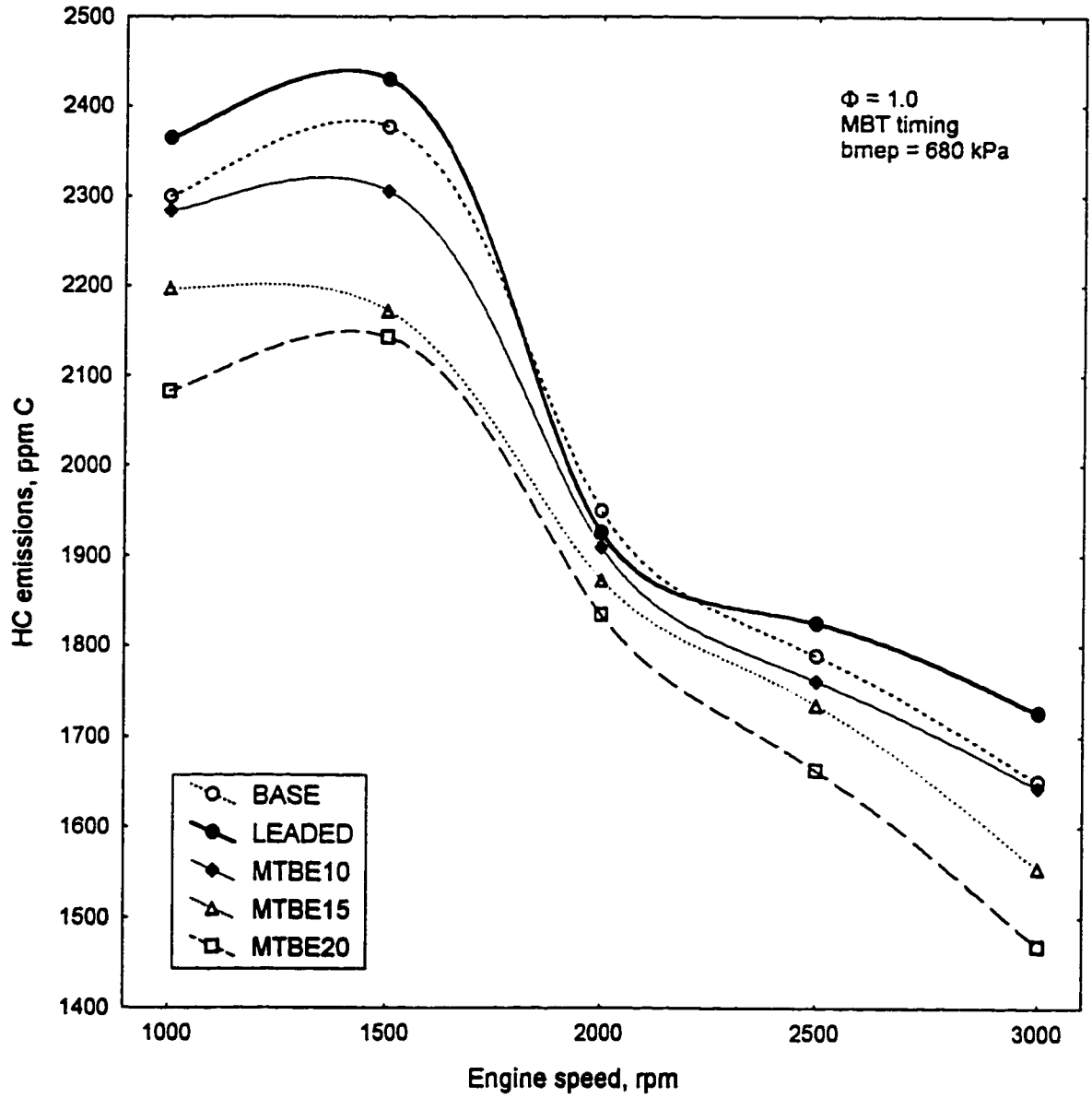

Figure 5-64: Exhaust HC emissions vs. speed at a constant load of 160 Nm (bmep = 680 kPa) for the MTBE blends.

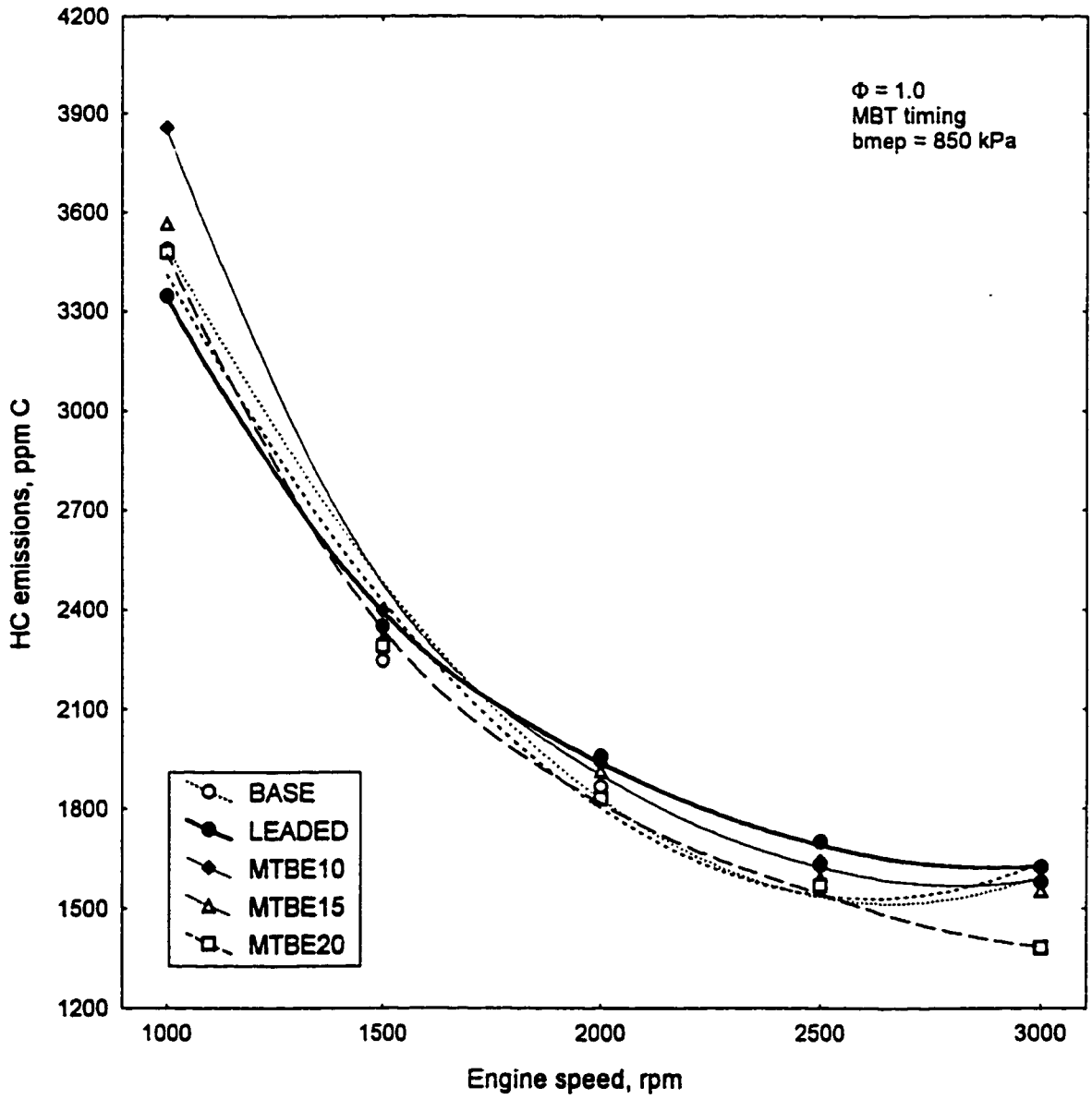

Figure 5-65: Exhaust HC emissions vs. speed at a constant load of 200 Nm (bmep = 850 kPa) for the MTBE blends.

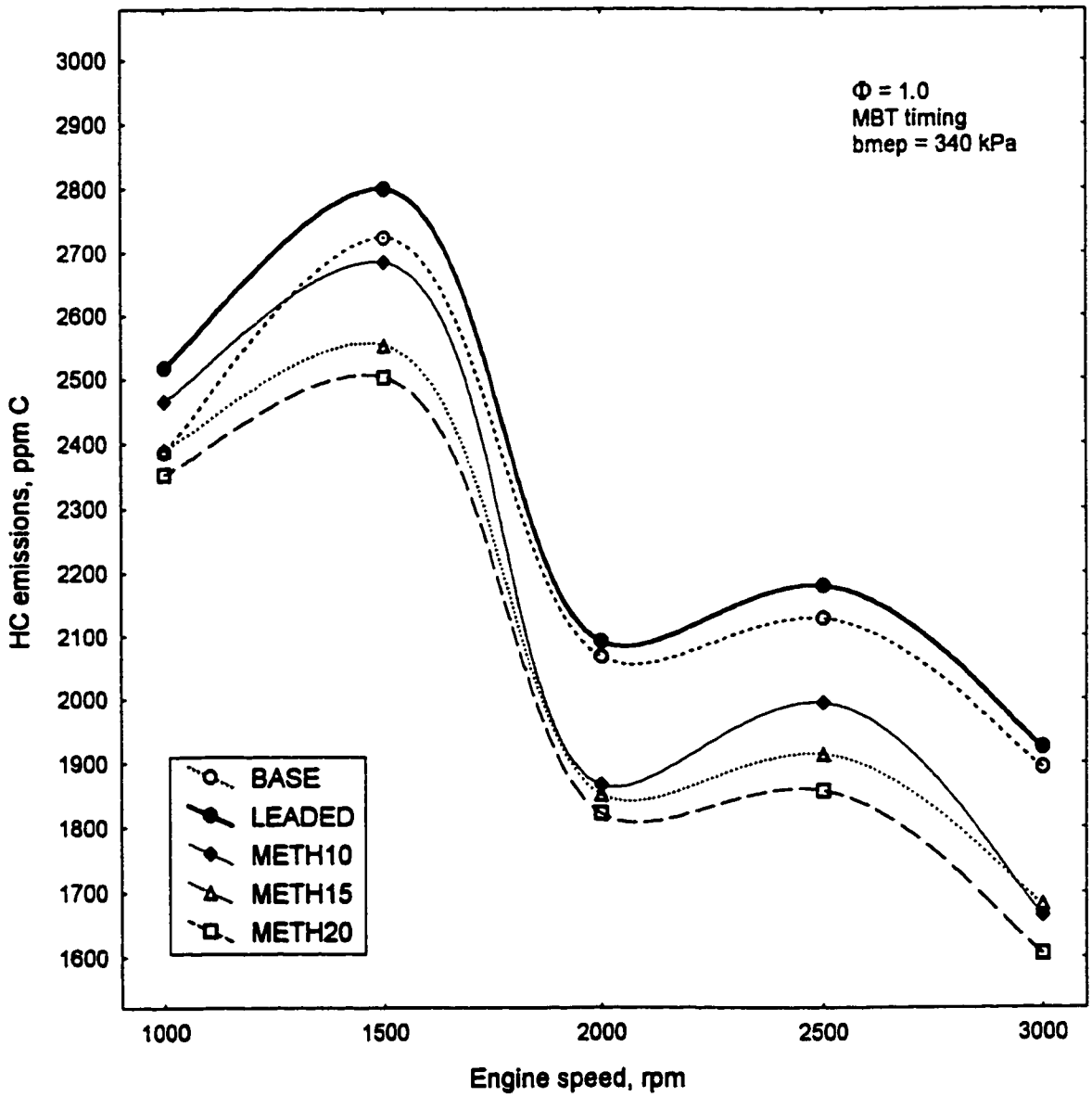

Figure 5-66: Exhaust HC emissions vs. speed at a constant load of 80 Nm (bmep = 340 kPa) for the methanol blends.

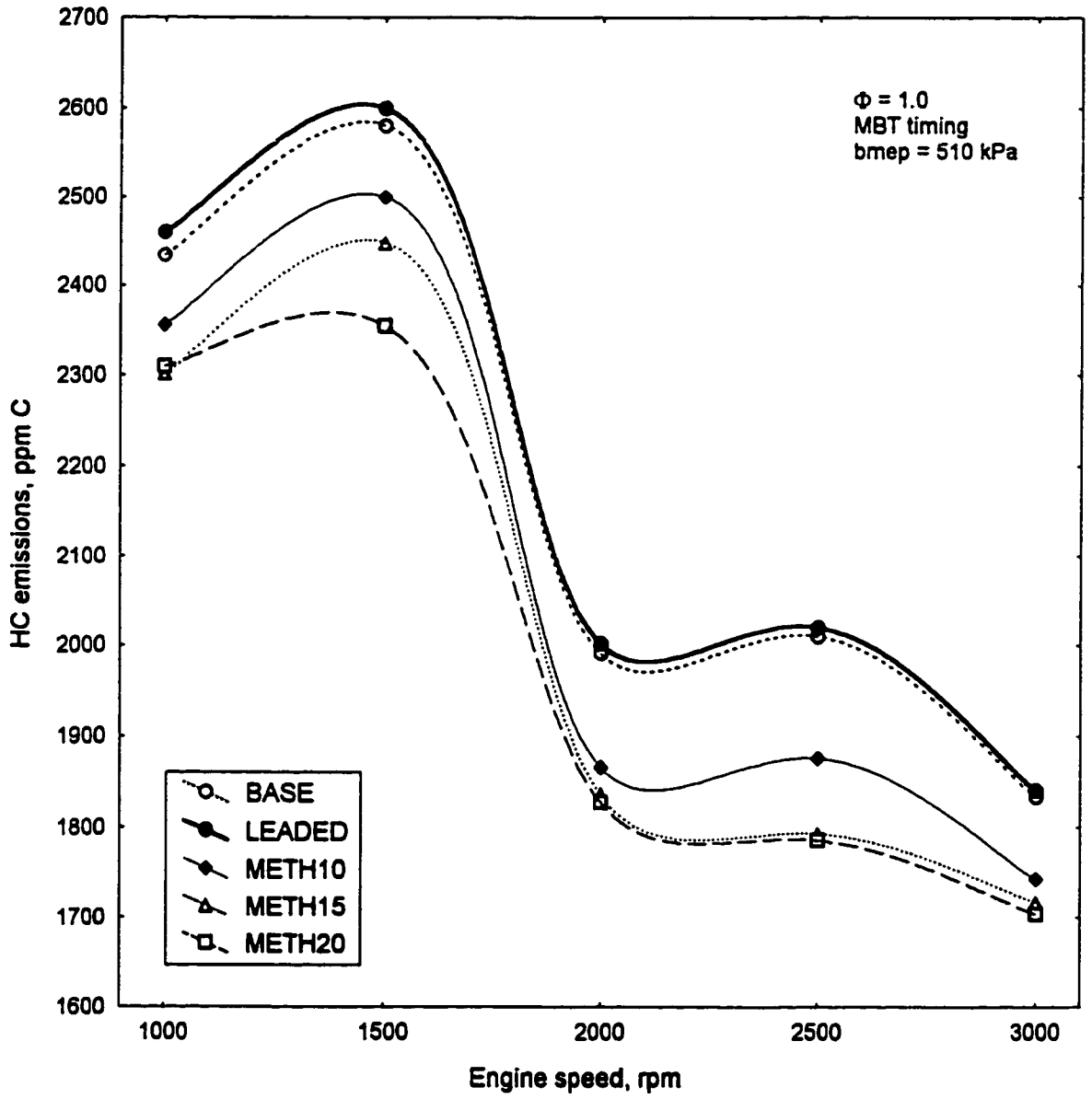

Figure 5-67: Exhaust HC emissions vs. speed at a constant load of 120 Nm (bmep = 510 kPa) for the methanol blends.

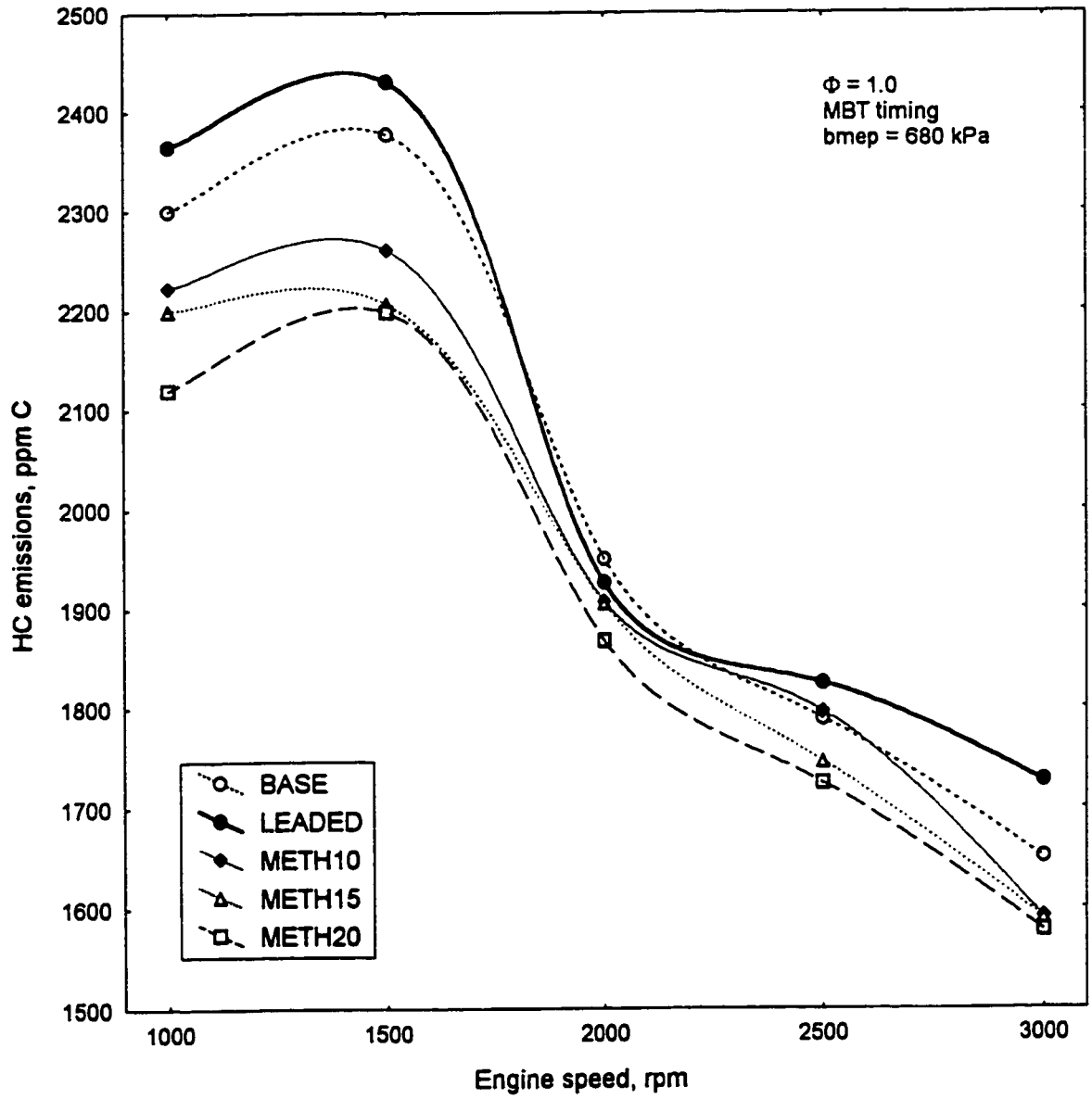

Figure 5-68: Exhaust HC emissions vs. speed at a constant load of 160 Nm (bmep = 680 kPa) for the methanol blends.

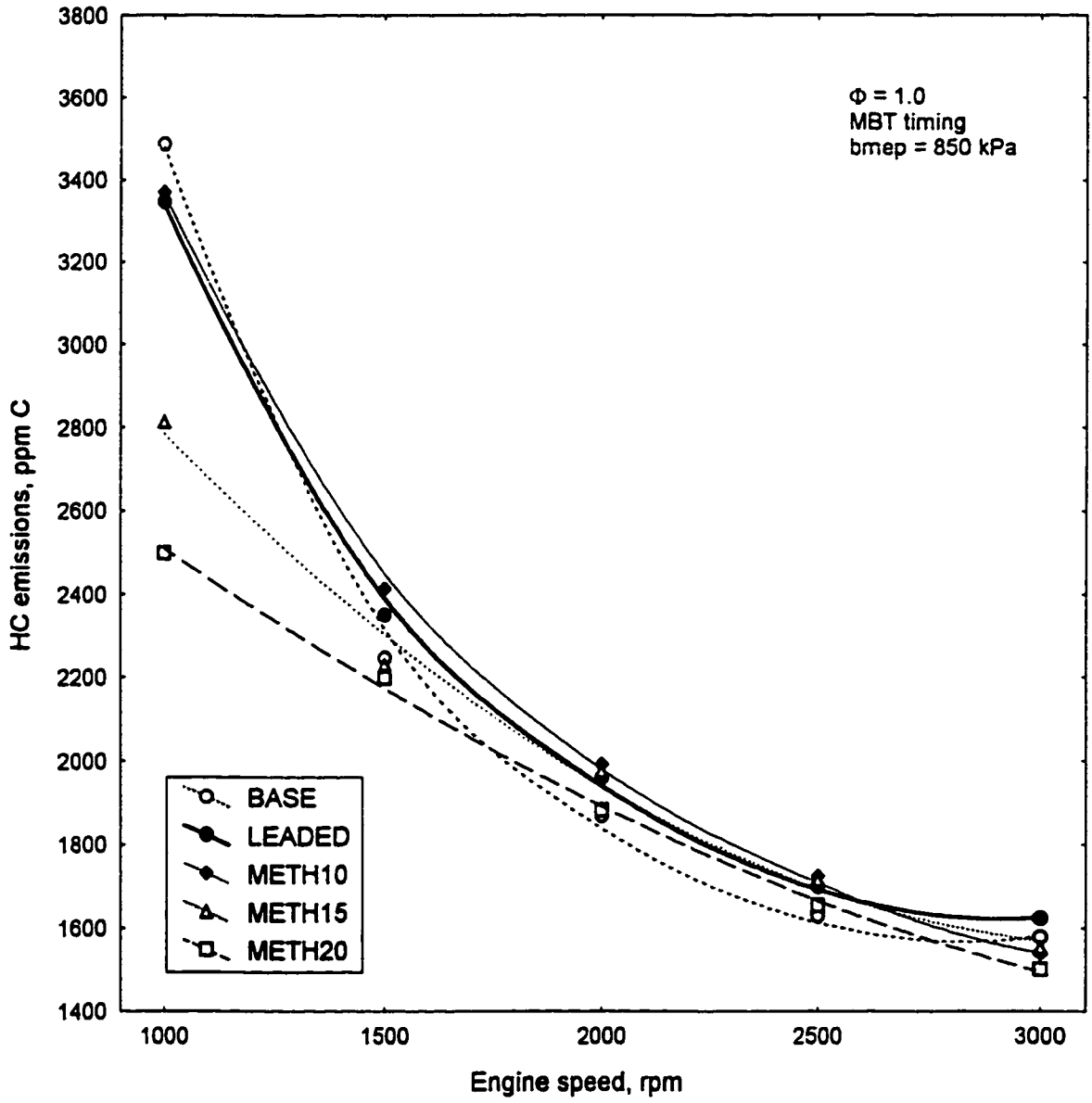

Figure 5-69: Exhaust HC emissions vs. speed at a constant load of 200 Nm (bmep = 850 kPa) for the methanol blends.

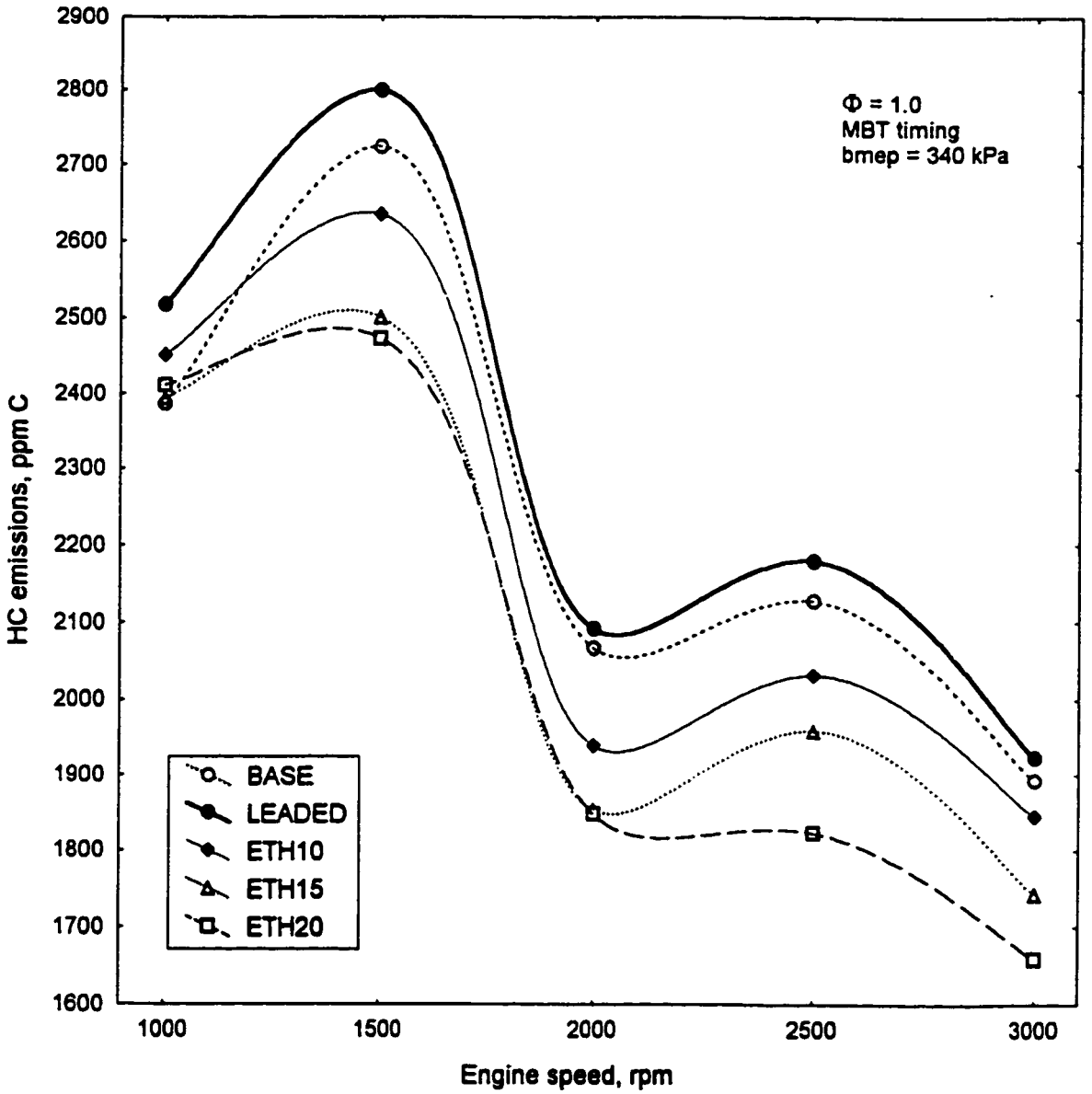

Figure 5-70: Exhaust HC emissions vs. speed at a constant load of 80 Nm (bmep = 340 kPa) for the ethanol blends.

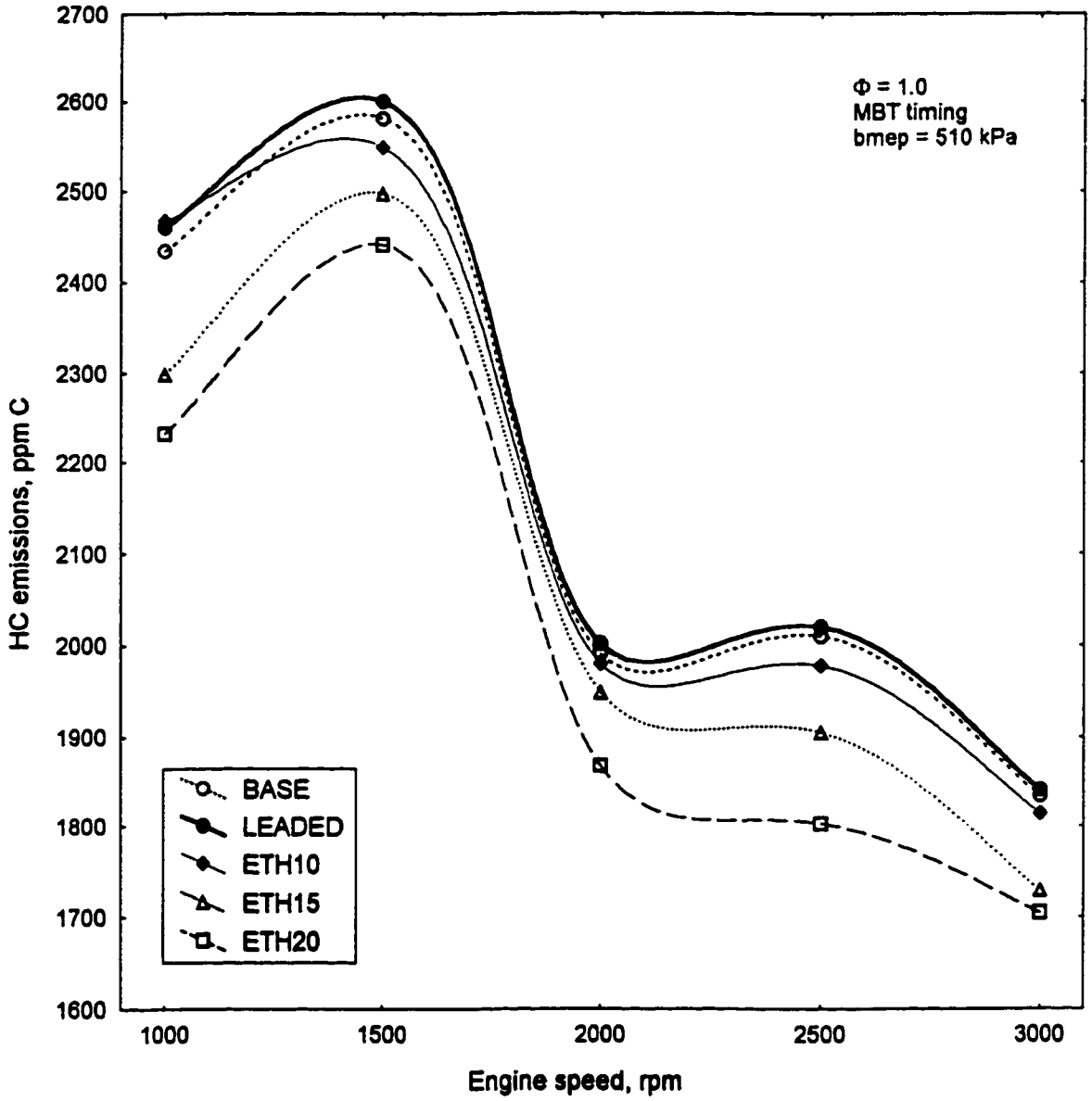

Figure 5-71: Exhaust HC emissions vs. speed at a constant load of 120 Nm (bmep = 510 kPa) for the ethanol blends.

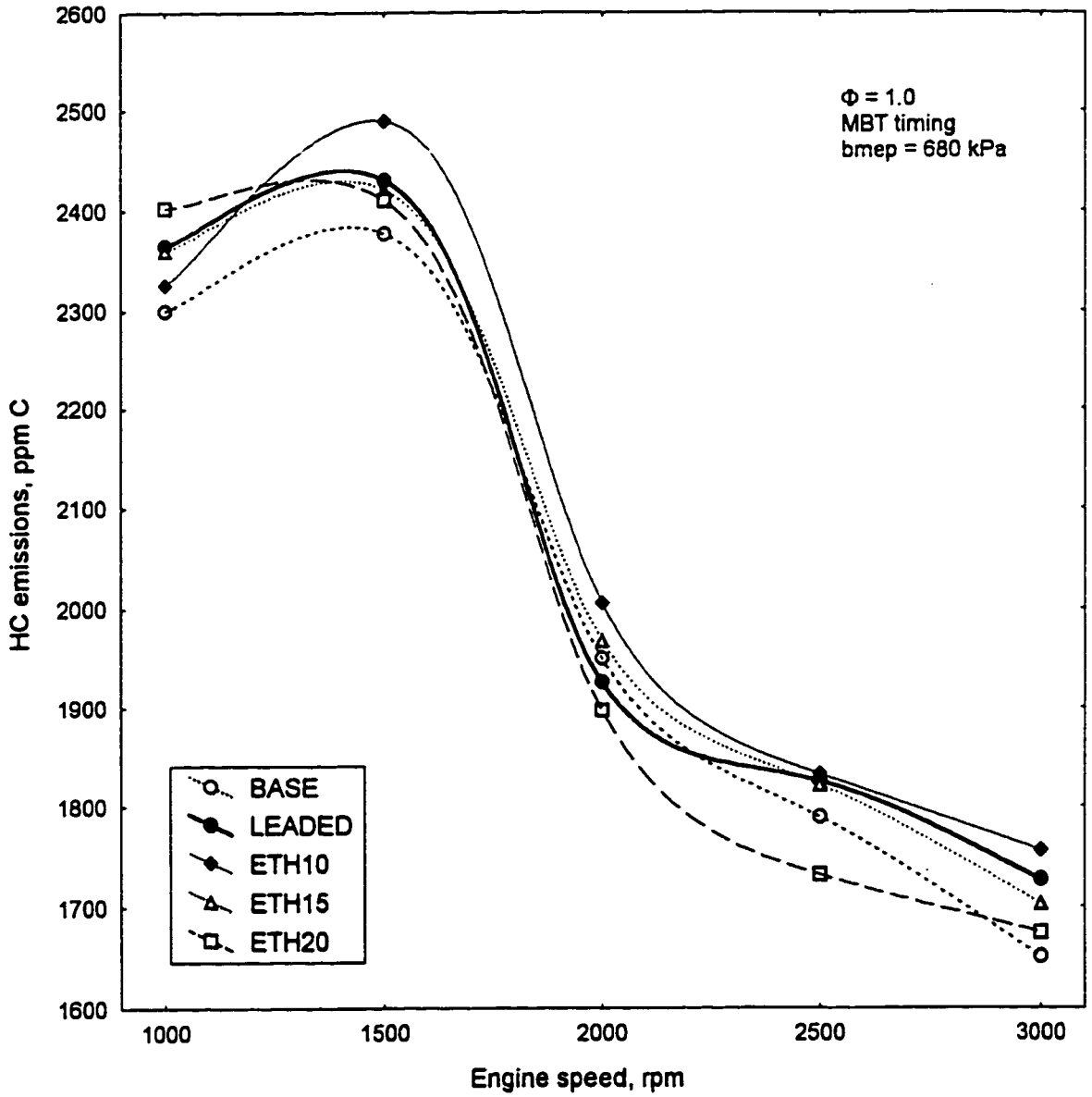

Figure 5-72: Exhaust HC emissions vs. speed at a constant load of 160 Nm (bmep = 680 kPa) for the ethanol blends.

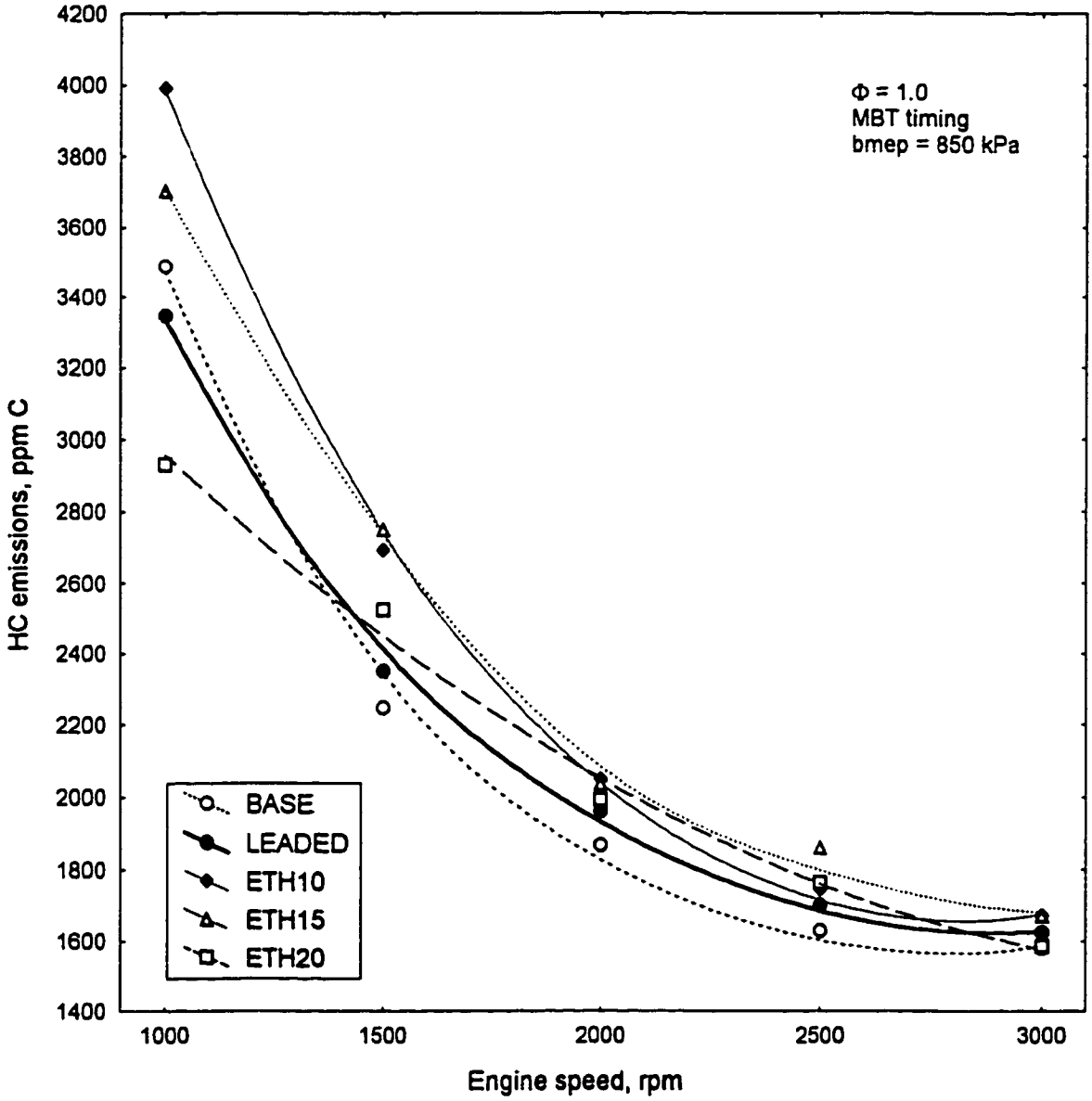

Figure 5-73: Exhaust HC emissions vs. speed at a constant load of 200 Nm (bmep = 850 kPa) for the ethanol blends.

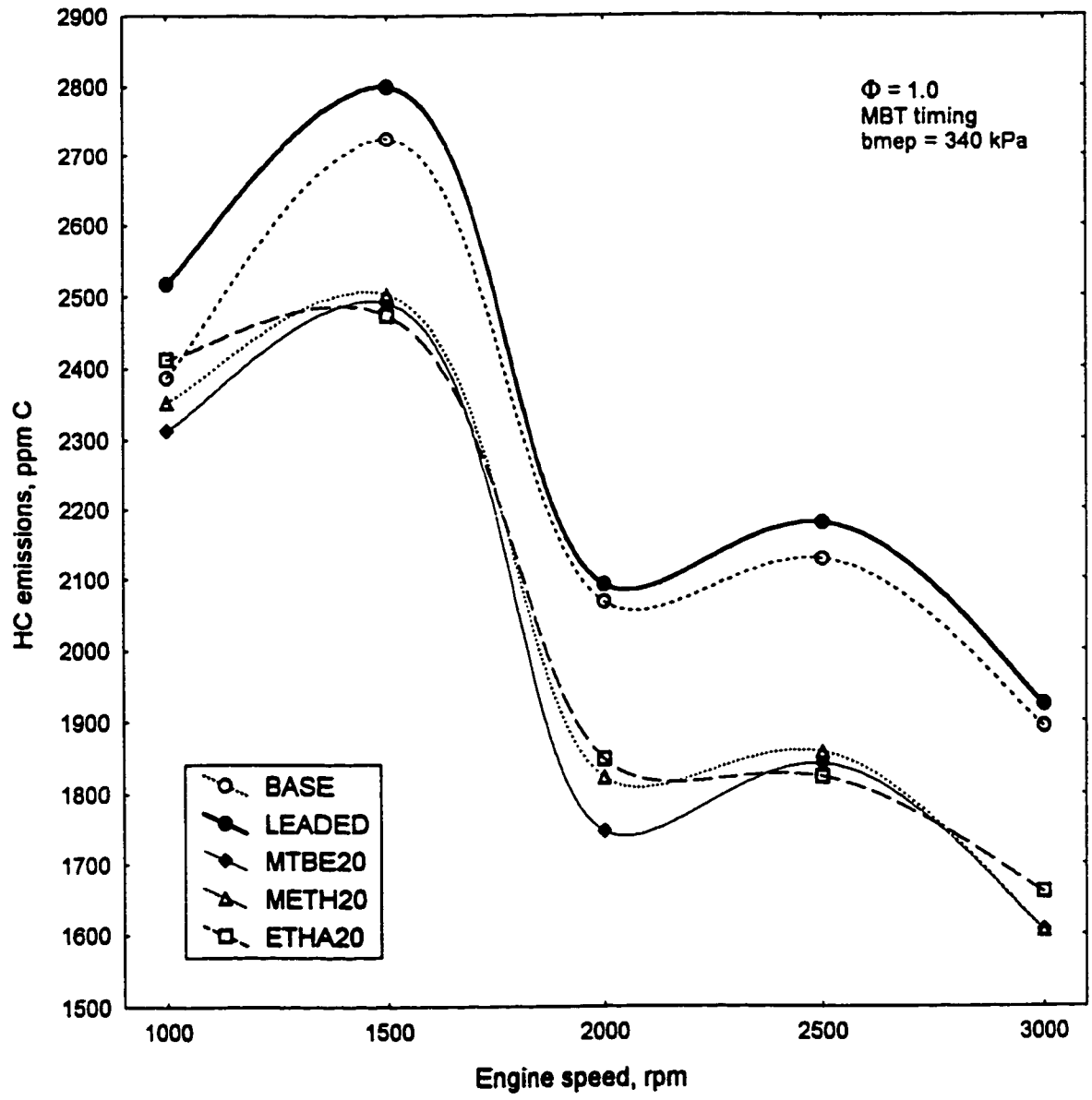

Figure 5-74: HC emissions vs. speed at a constant load of 80 Nm (bmep = 340 kPa) for the least emitting blends compared to the base and leaded fuels.

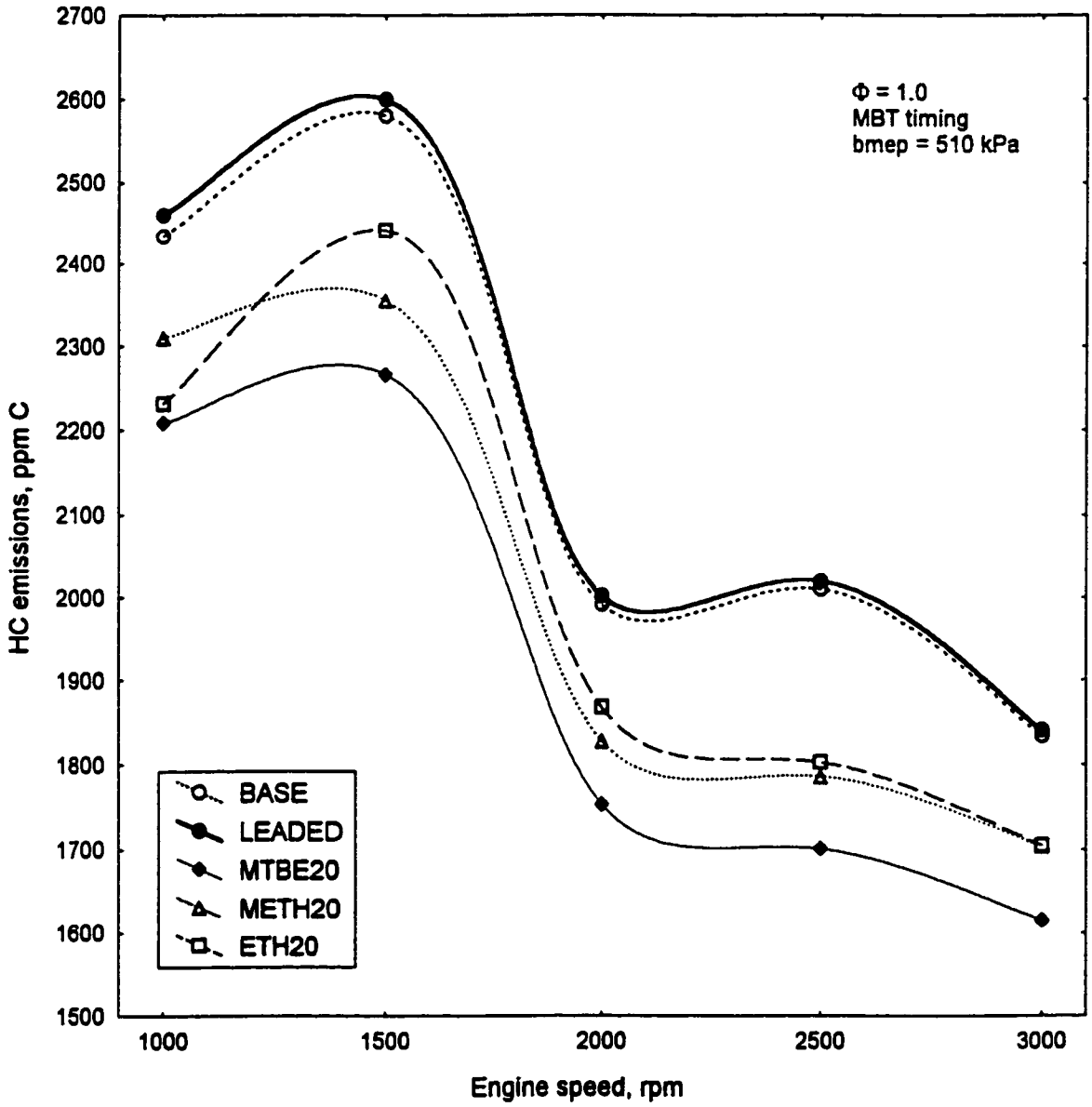

Figure 5-75: HC emissions vs. speed at a constant load of 120 Nm (bmep = 510 kPa) for the least emitting blends compared to the base and leaded fuels.

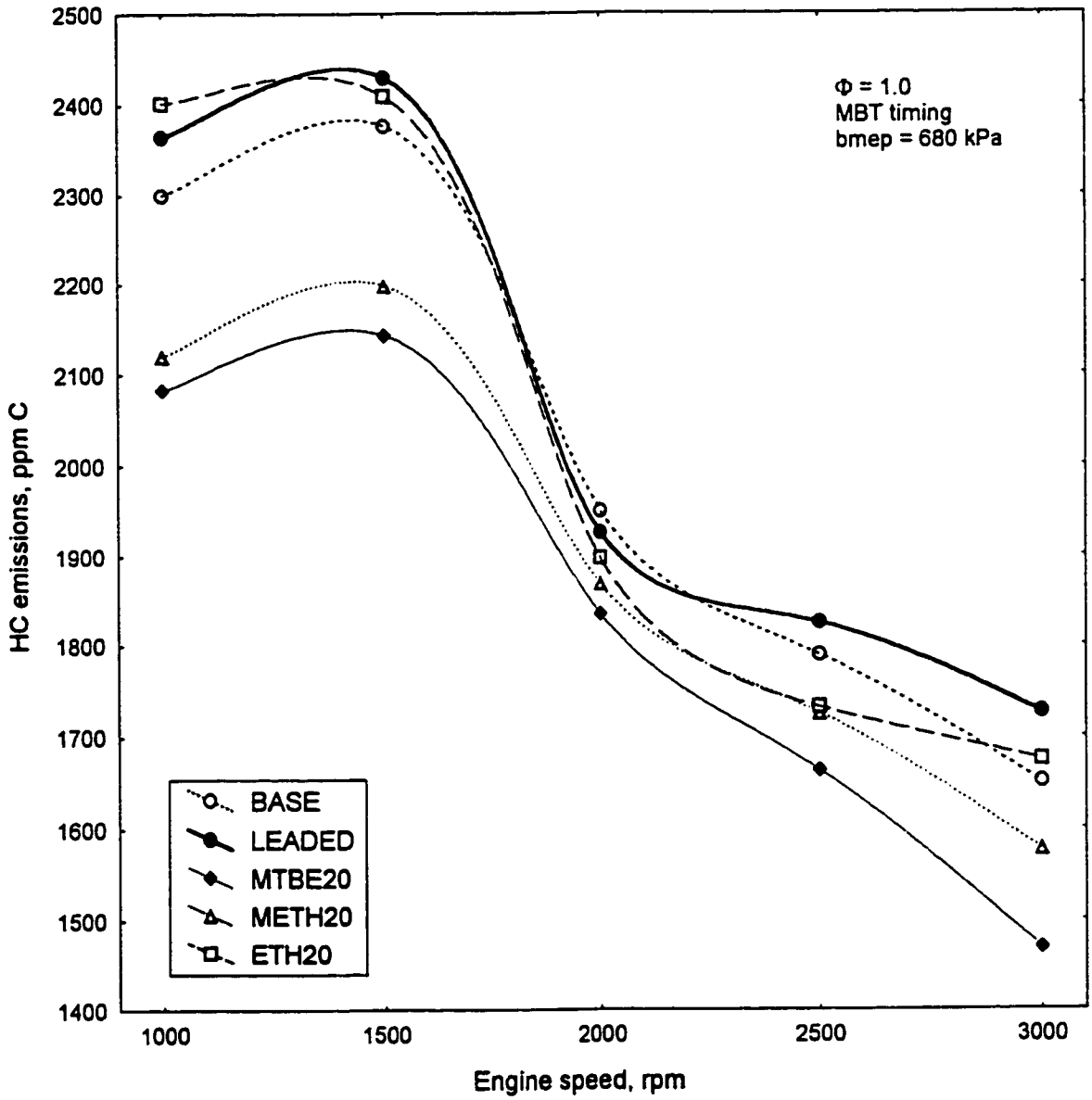

Figure 5-76: HC emissions vs. speed at a constant load of 160 Nm (bmep = 680 kPa) for the least emitting blends compared to the base and leaded fuels.

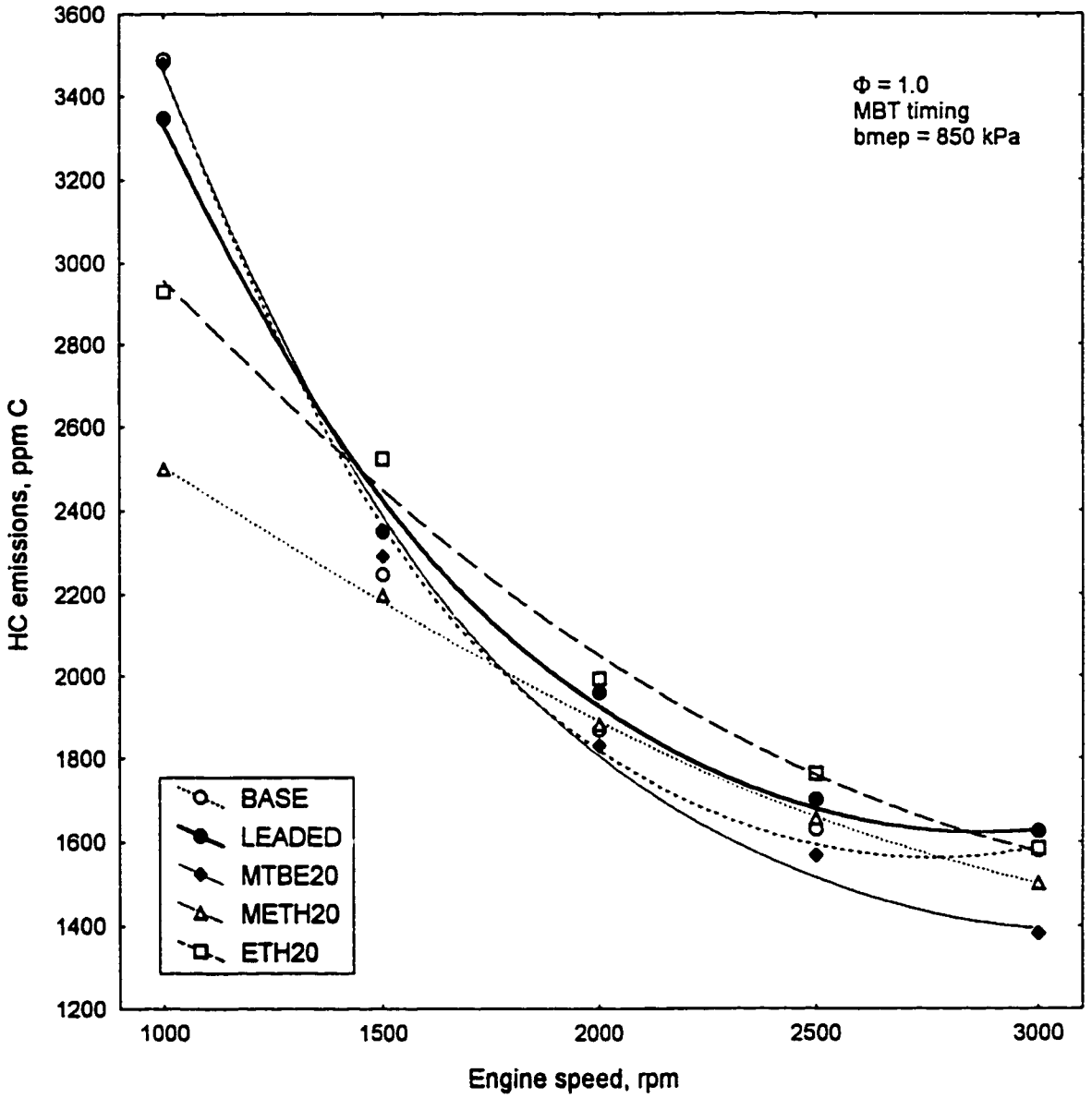

Figure 5-77: HC emissions vs. speed at a constant load of 200 Nm (bmep = 850 kPa) for the least emitting blends compared to the base and leaded fuels.

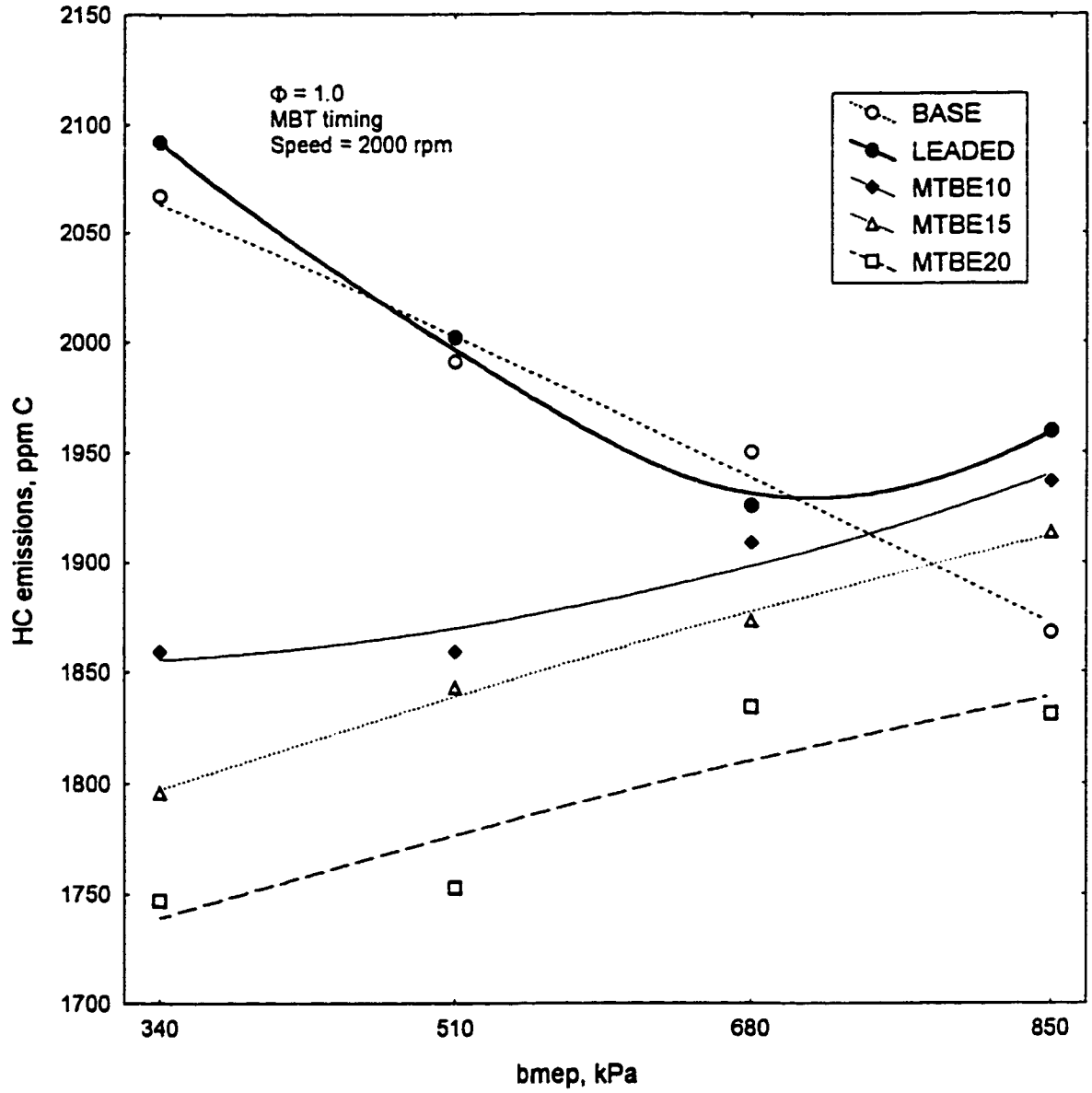

Figure 5-78: Effect of load on HC emissions at a constant speed of 2000 rpm for the MTBE blends.

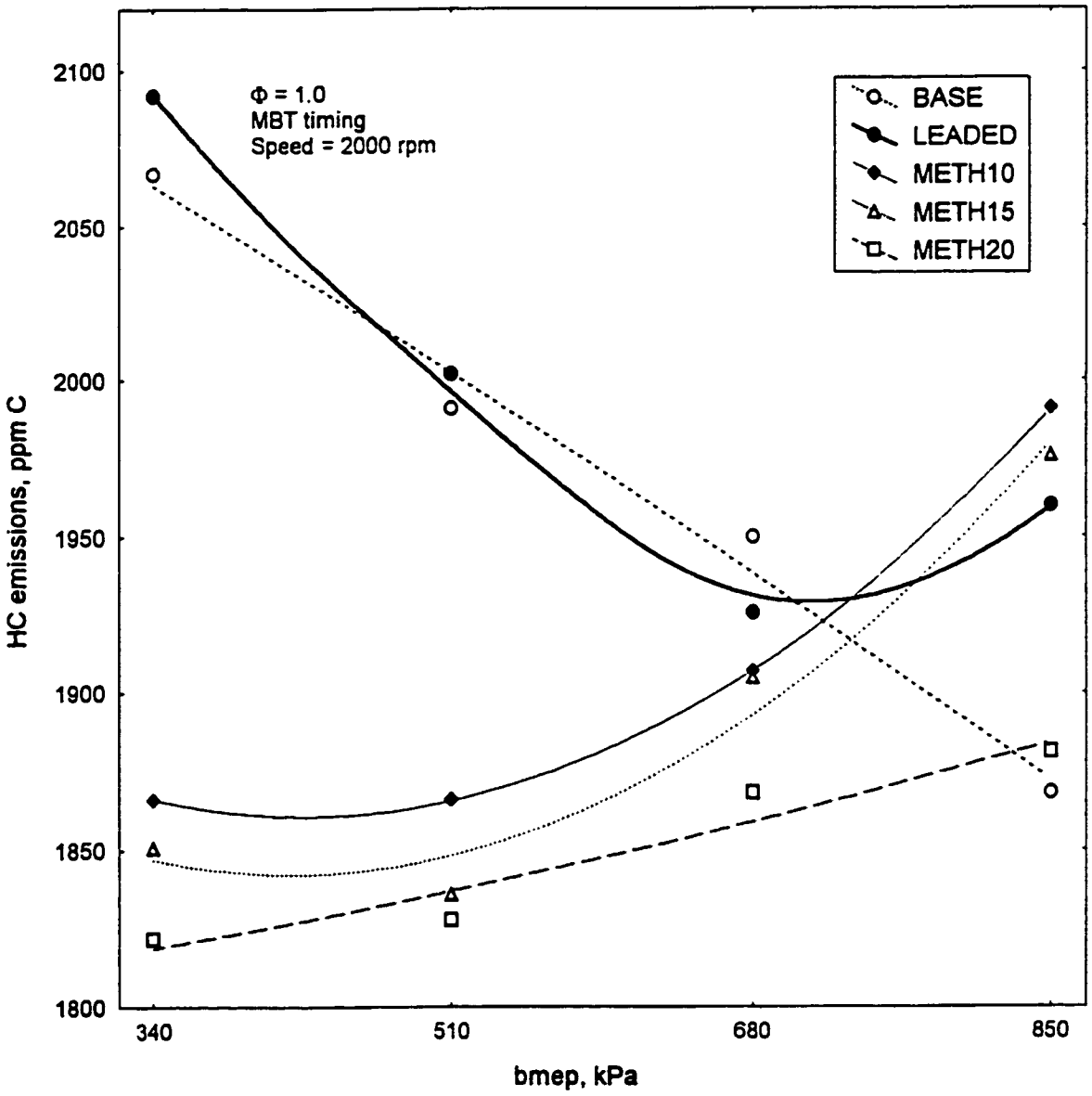

Figure 5-79: Effect of load on HC emissions at a constant speed of 2000 rpm for the methanol blends.

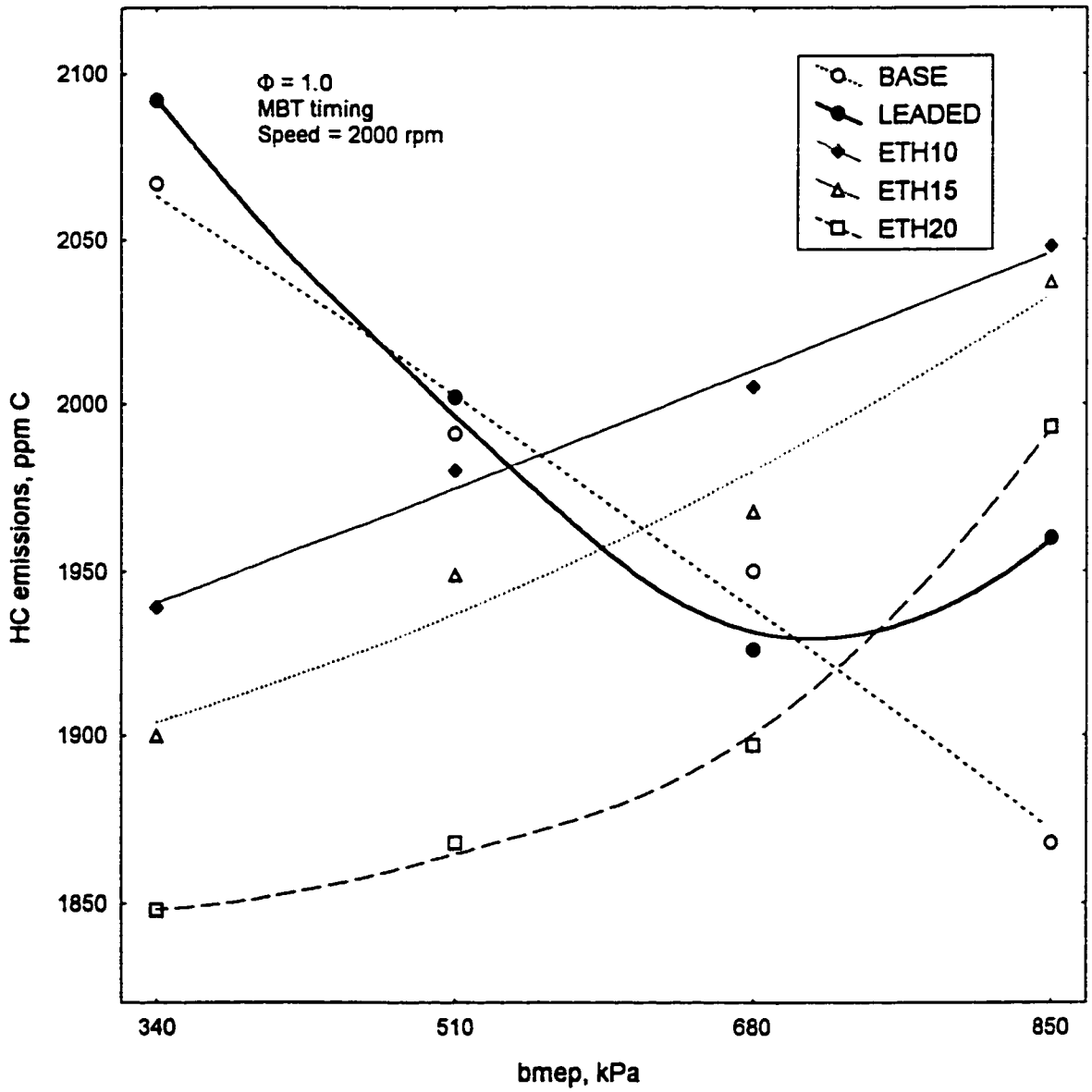

Figure 5-80: Effect of load on HC emissions at a constant speed of 2000 rpm for the ethanol blends.

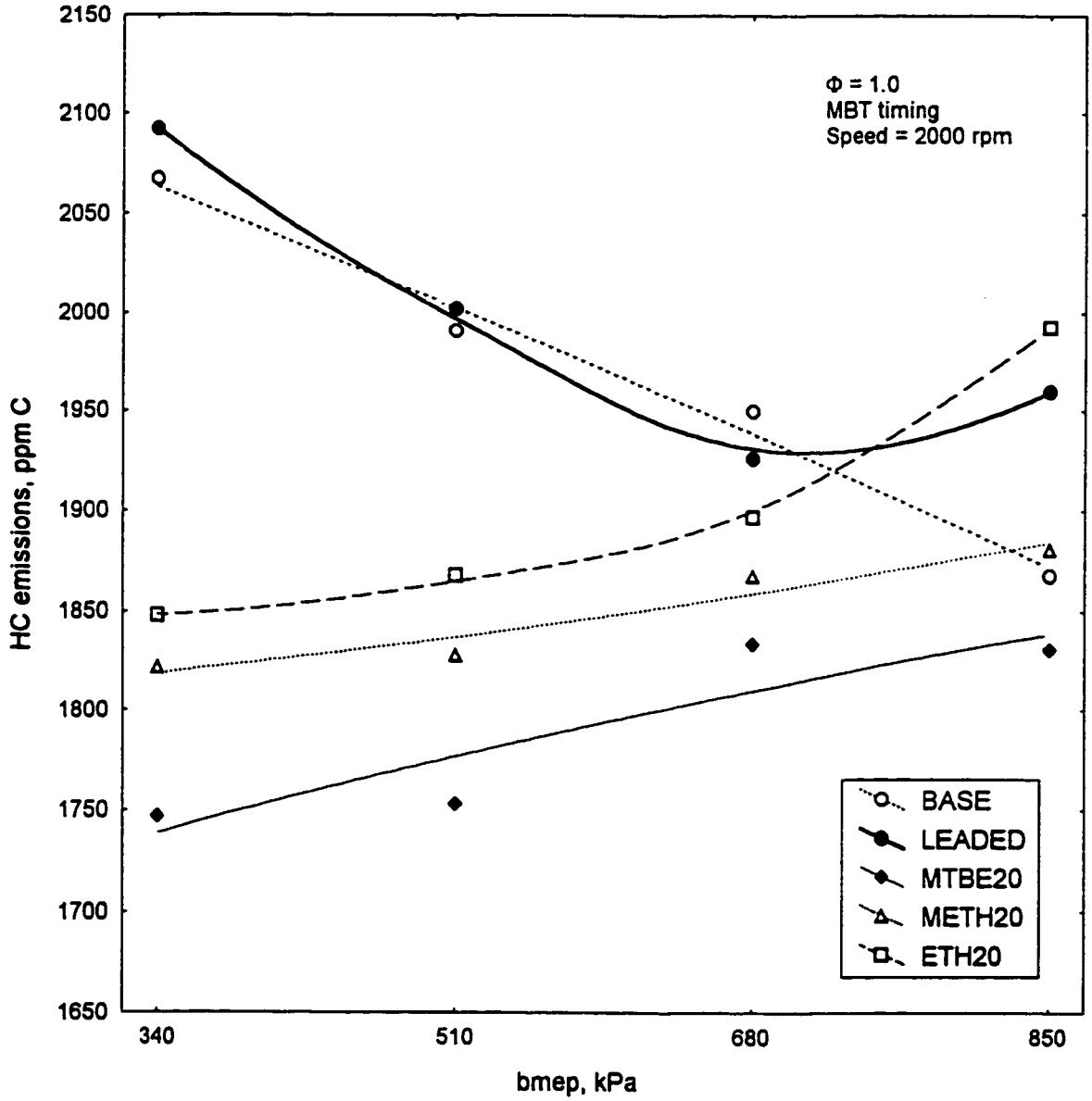

Figure 5-81: Effect of load on HC emissions at a constant speed of 2000 rpm for the least emitting blends compared to the base and leaded fuels.

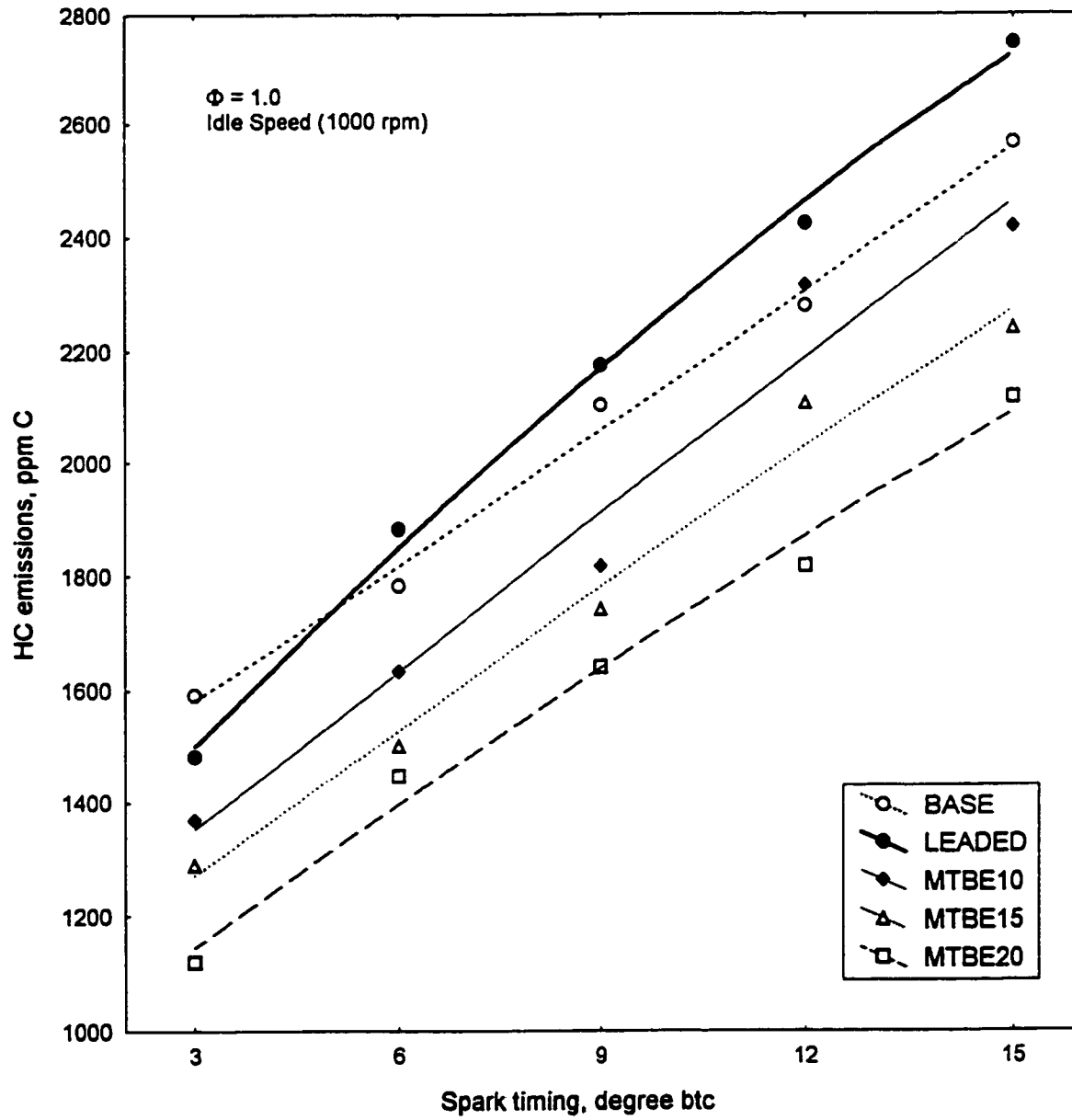

Figure 5-82: Effect of spark timing on HC emissions at idle speed for the MTBE blends.

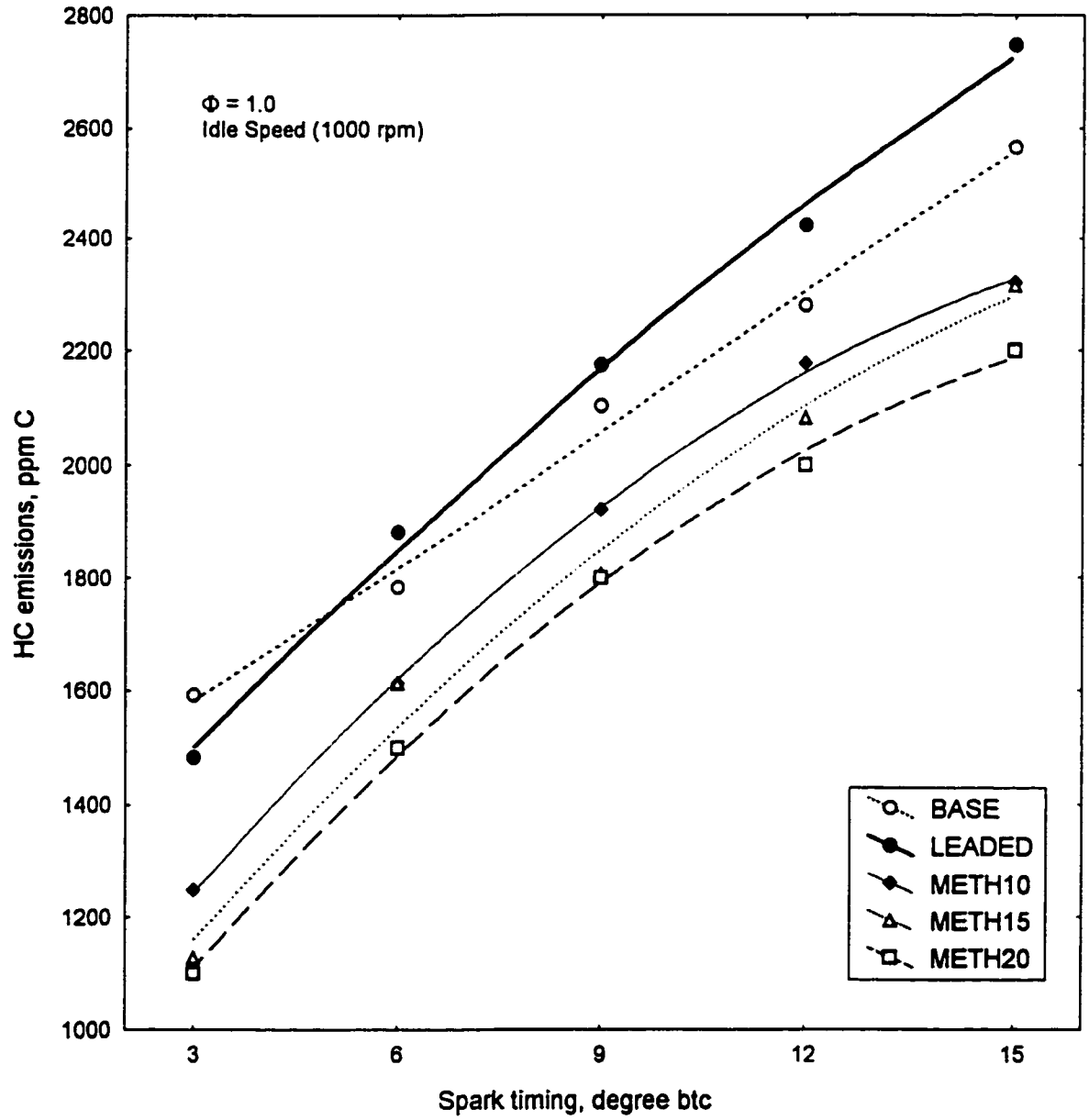

Figure 5-83: Effect of spark timing on HC emissions at idle speed for the methanol blends.

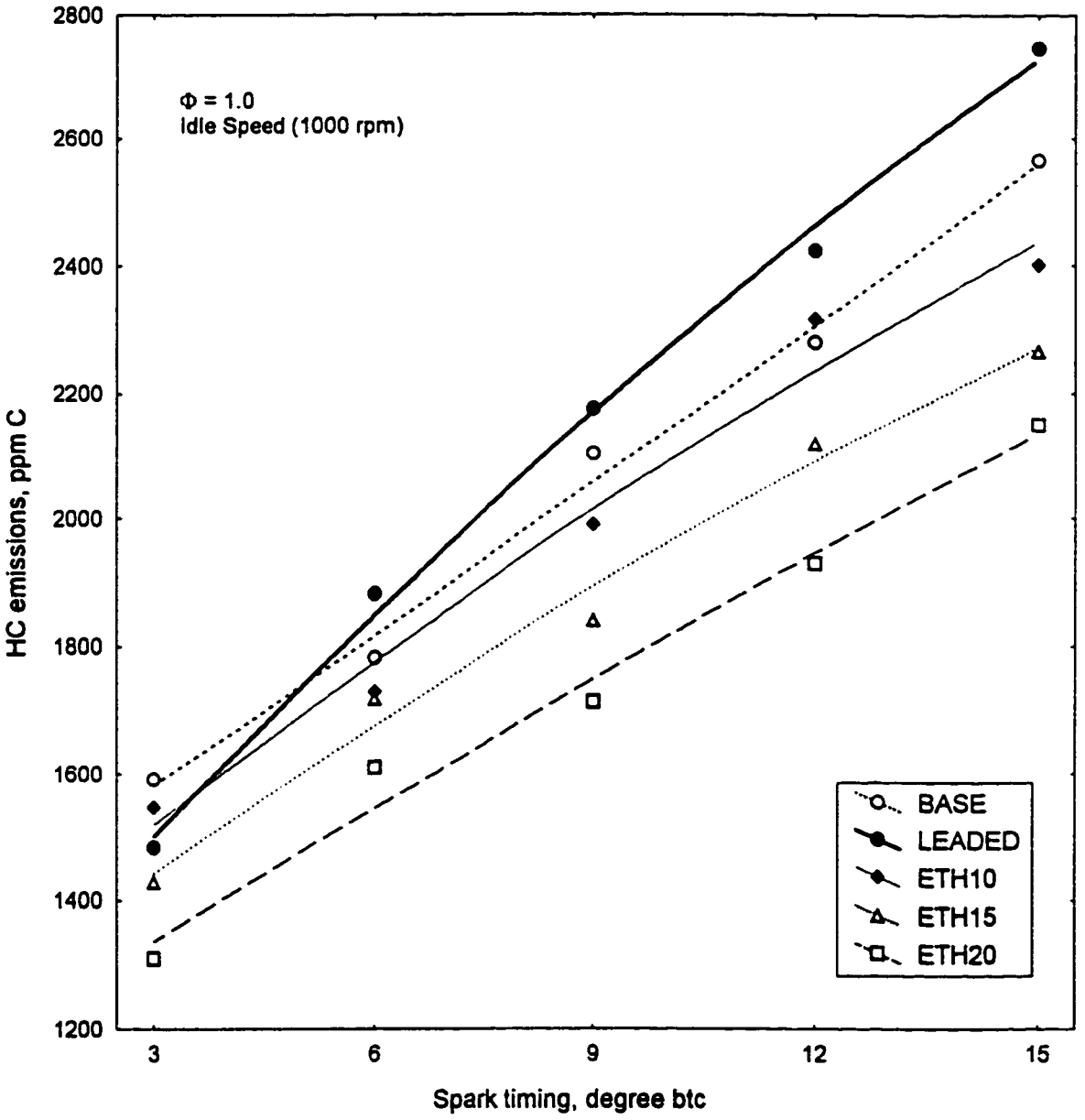

Figure 5-84: Effect of spark timing on HC emissions at idle speed for the ethanol blends.

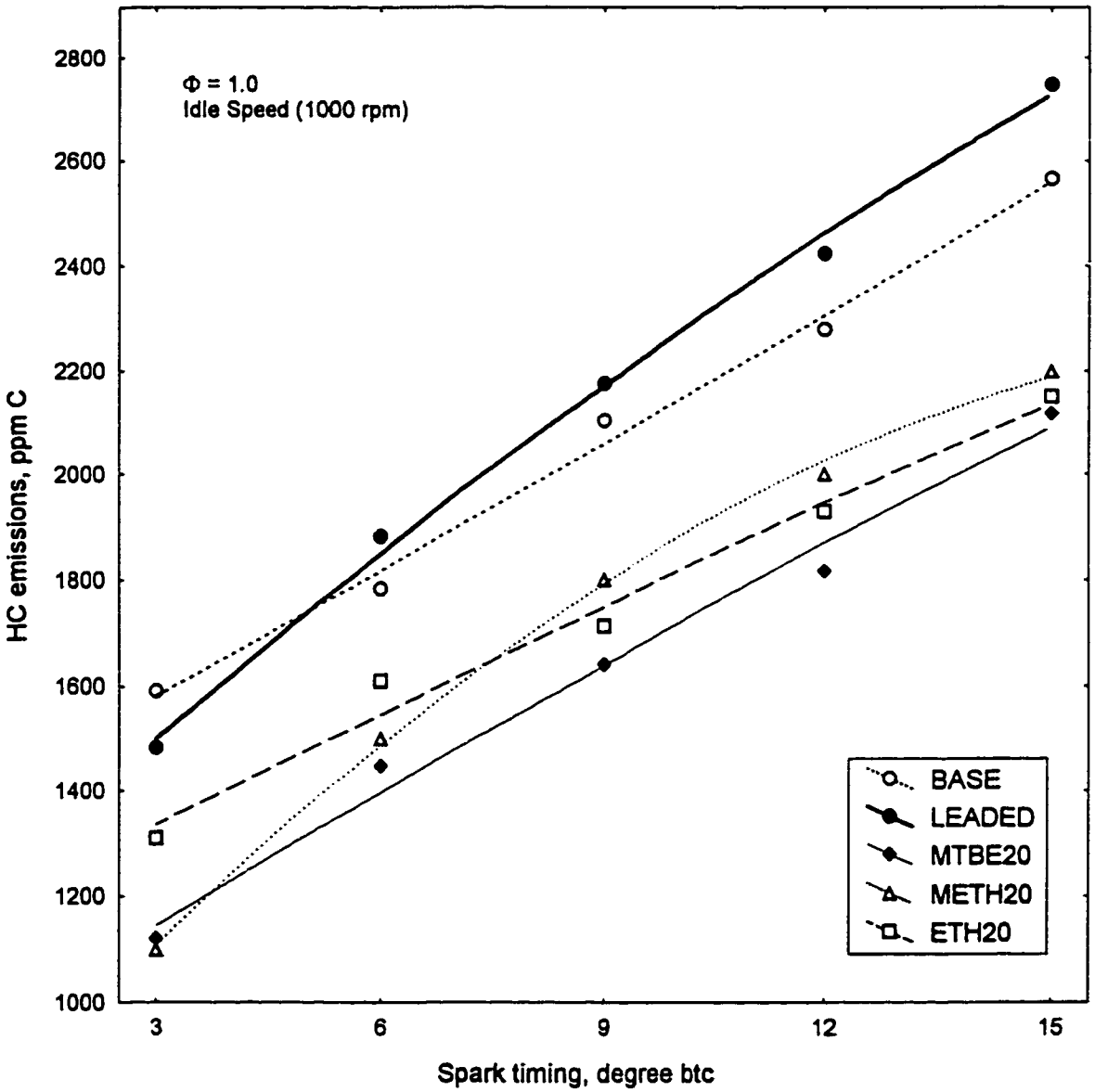

Figure 5-85: Effect of spark timing on HC emissions at idle speed for the least emitting blends compared to the base and leaded fuels.

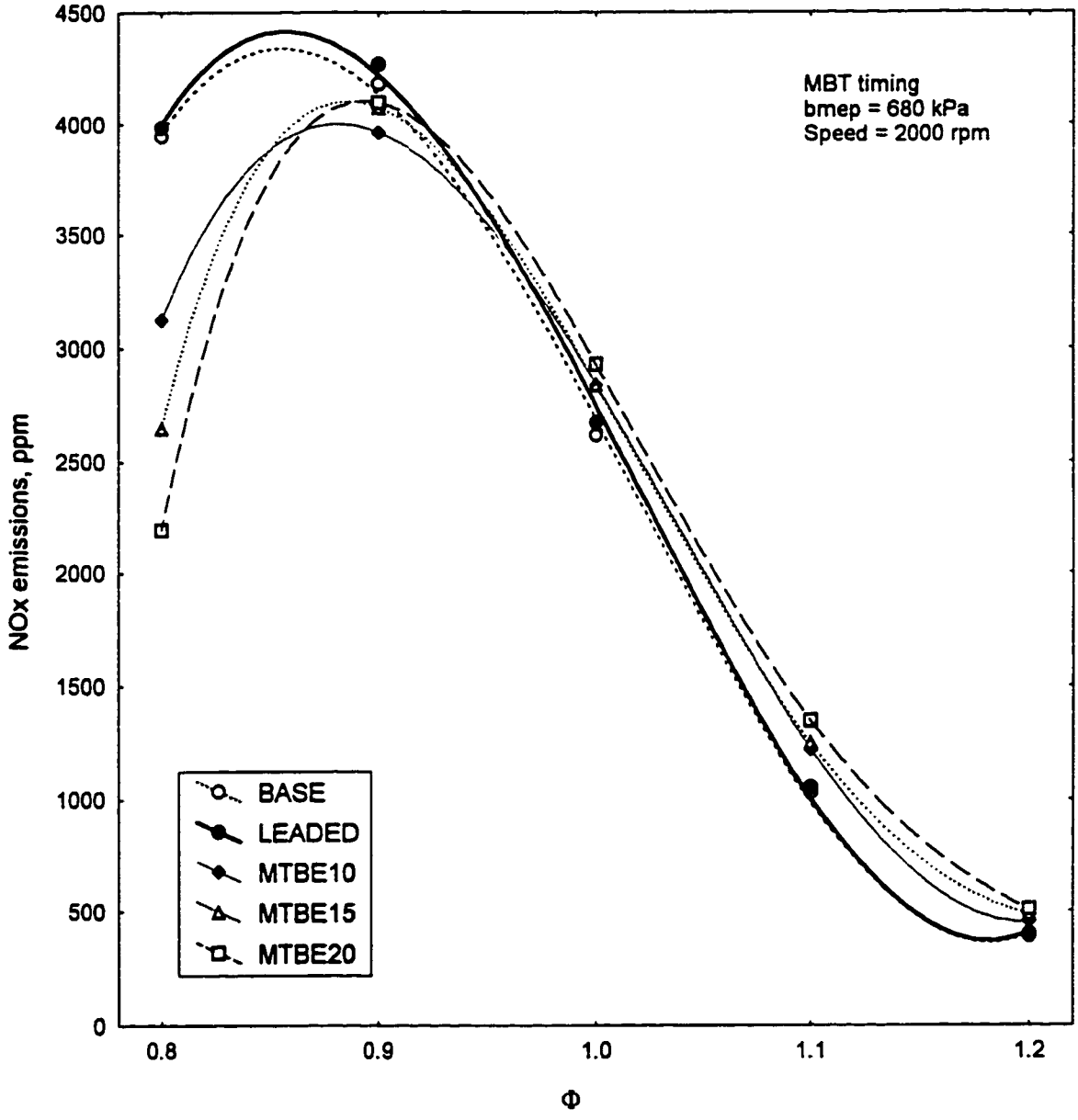

Figure 5-86: Effect of equivalence ratio on NOx emissions for the MTBE blends.

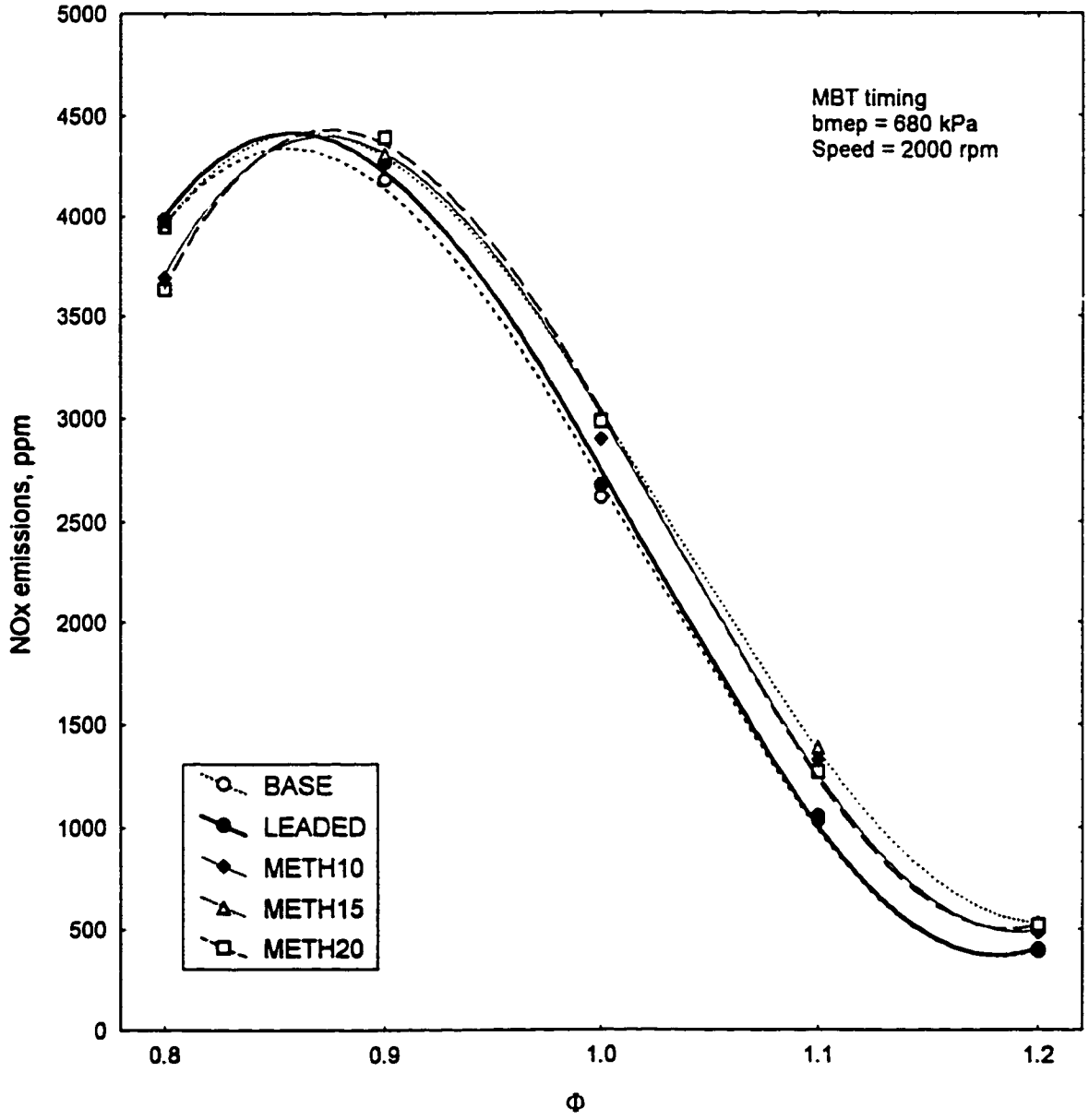

Figure 5-87: Effect of equivalence ratio on NOx emissions for the methanol blends.

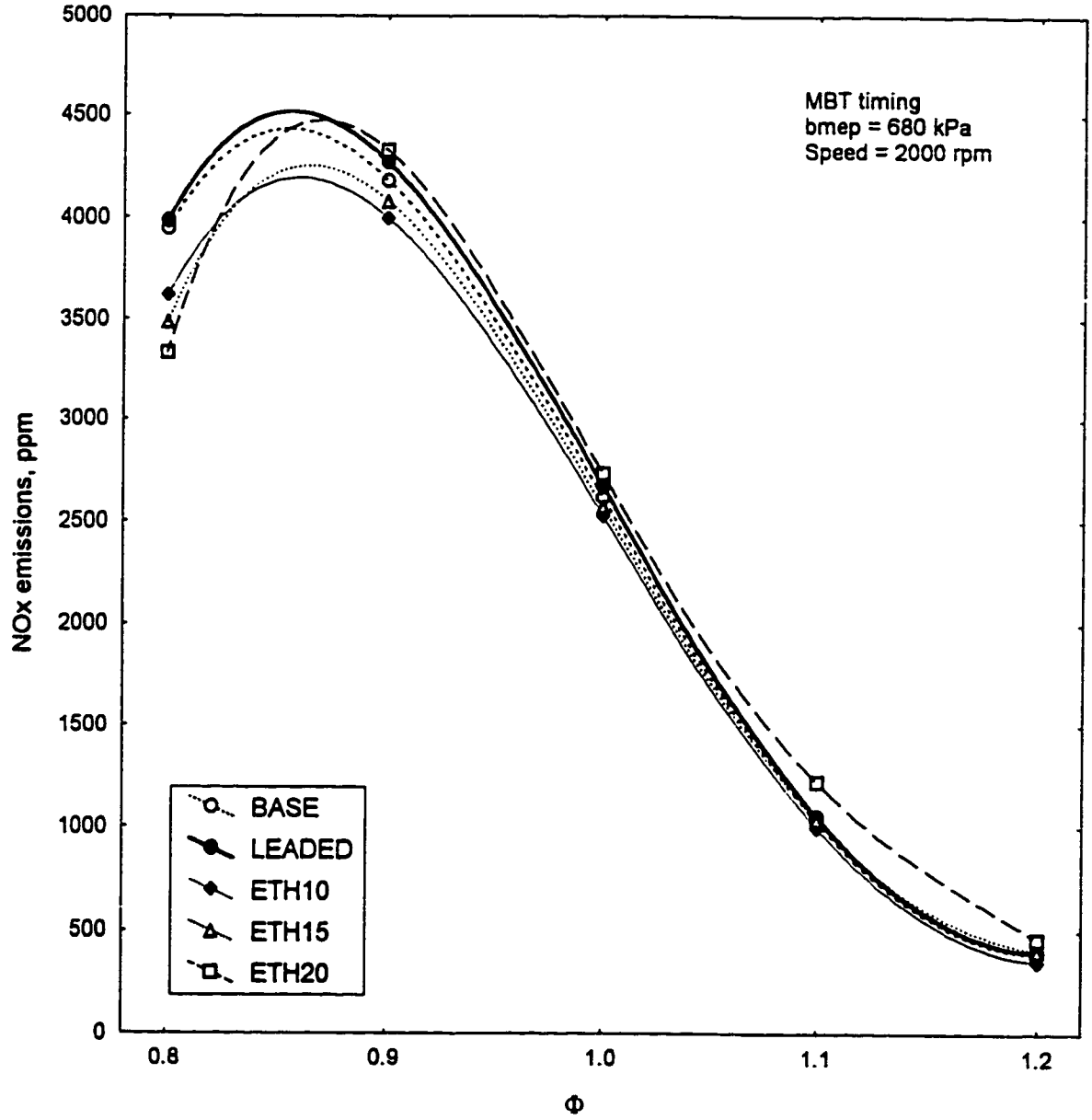

Figure 5-88: Effect of equivalence ratio on NOx emissions for the ethanol blends.

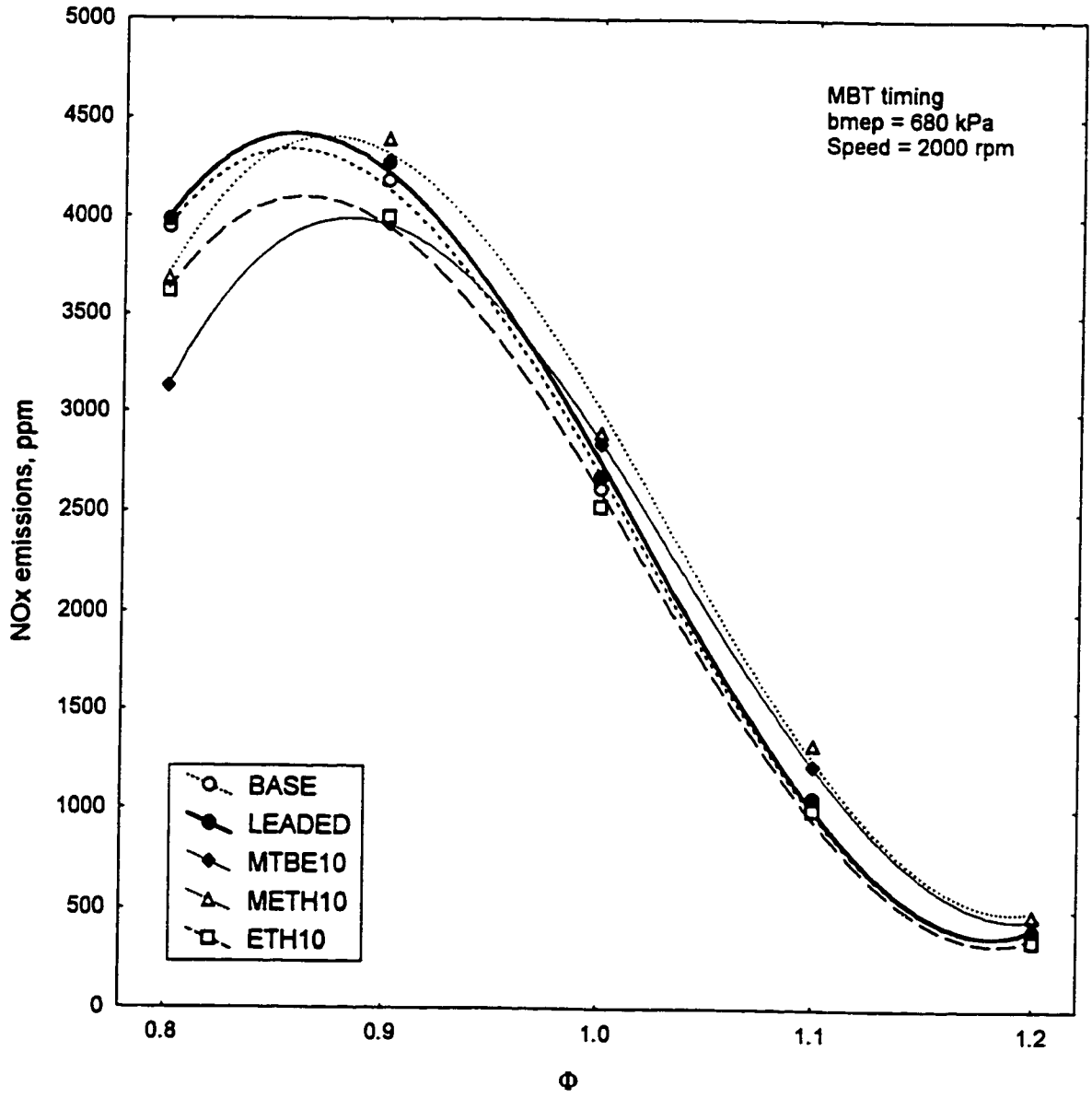

Figure 5-89: Effect of equivalence ratio on NOx emissions for the least emitting blends compared to the base and leaded fuels.

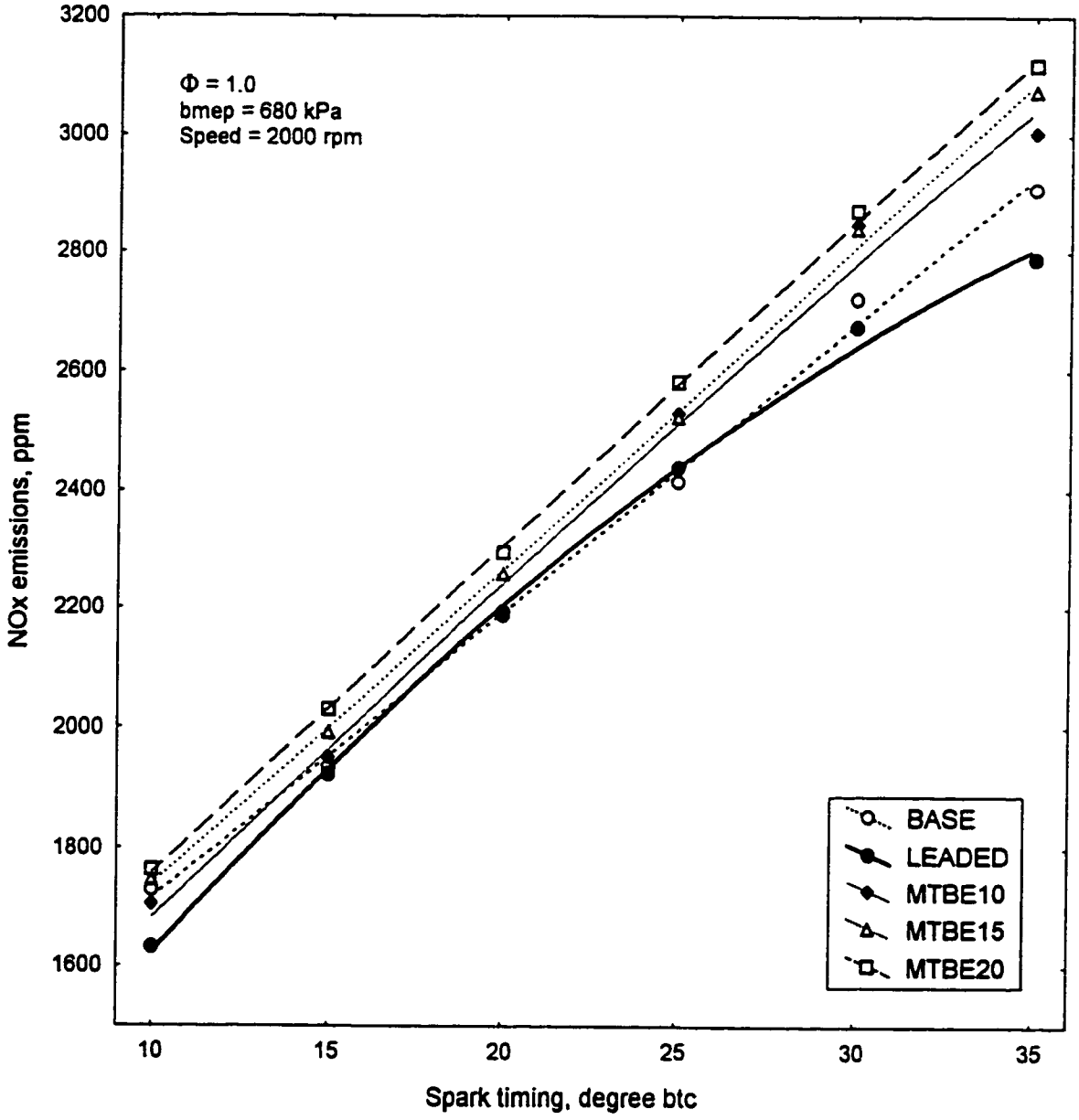

Figure 5-90: Effect of spark timing on NOx emissions for the MTBE blends.

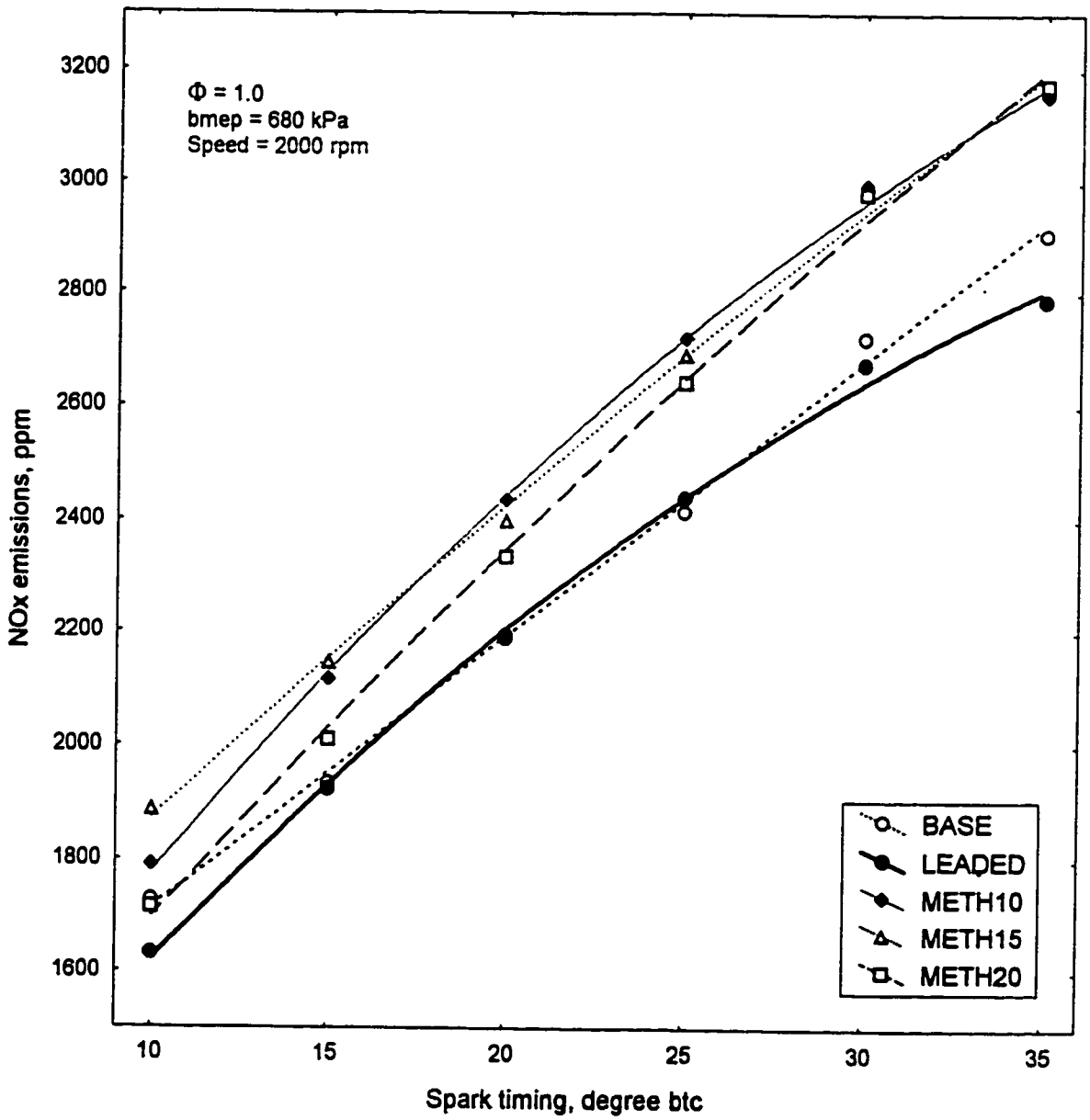

Figure 5-91: Effect of spark timing on NOx emissions for the methanol blends.

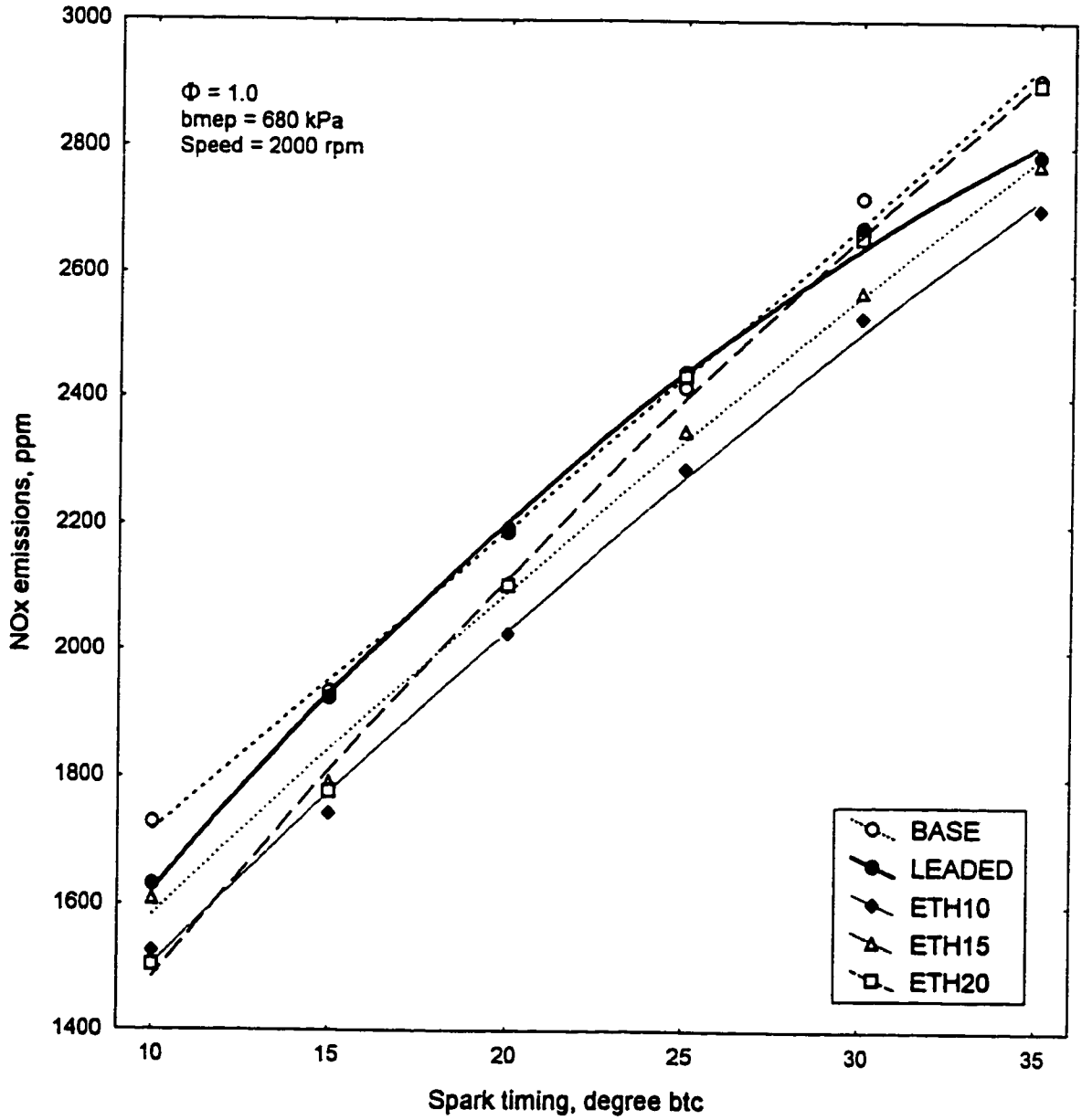

Figure 5-92: Effect of spark timing on NOx emissions for the ethanol blends.

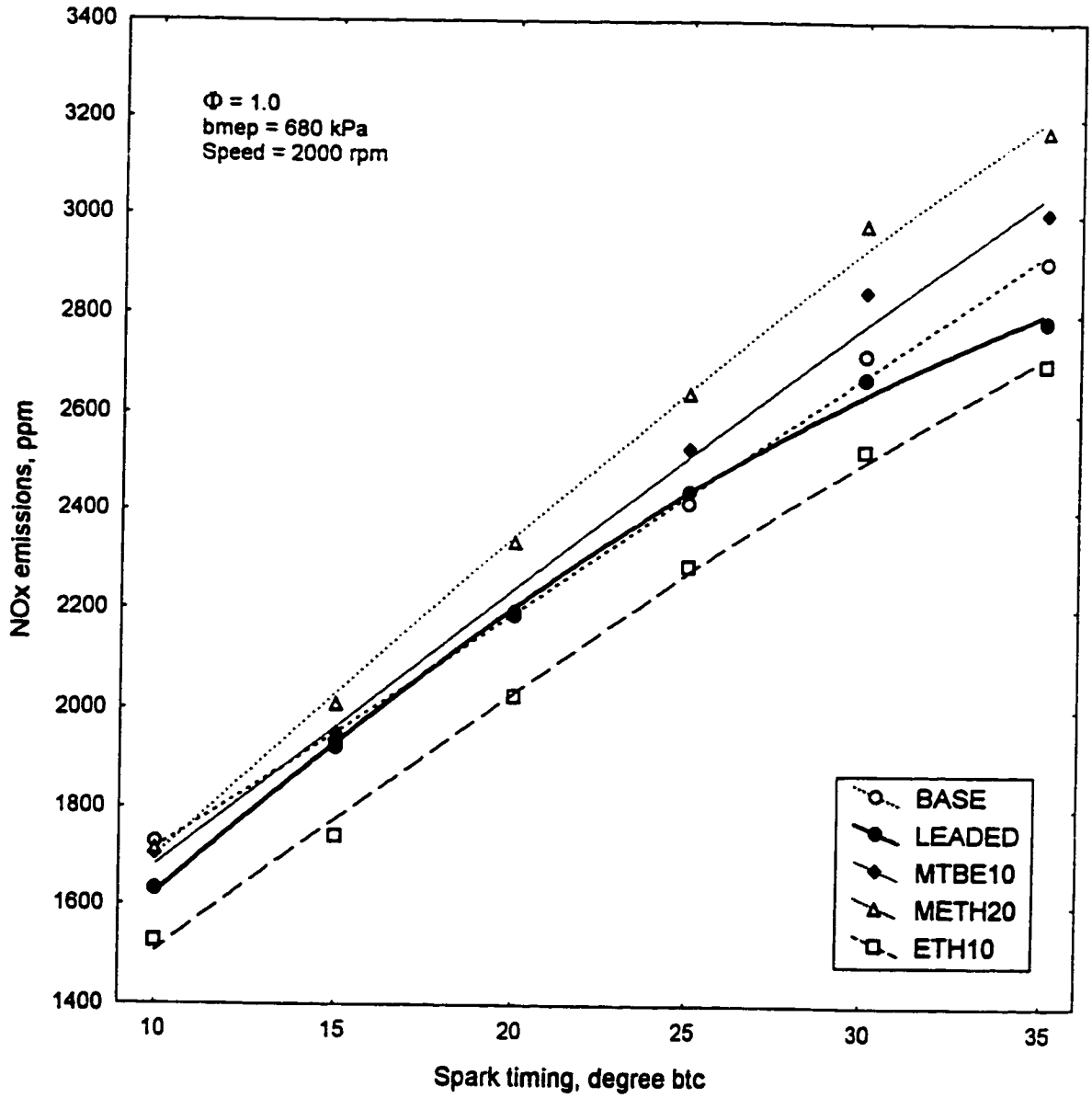

Figure 5-93: Effect of spark timing on NOx emissions for the least emitting blends compared to the base and leaded fuels.

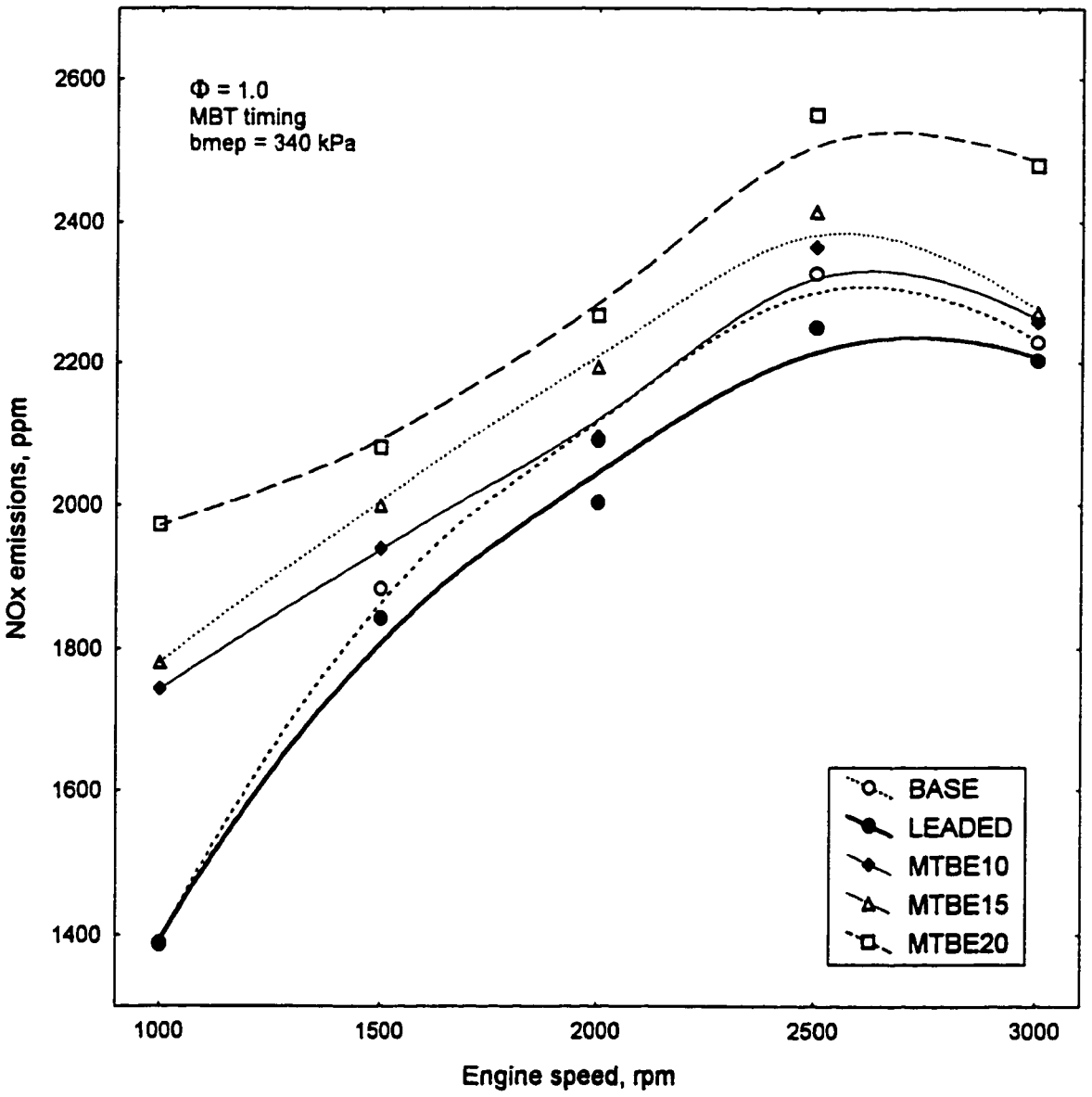

Figure 5-94: NOx emissions vs. speed at a constant load of 80 Nm (bmep = 340 kPa) for the MTBE blends.

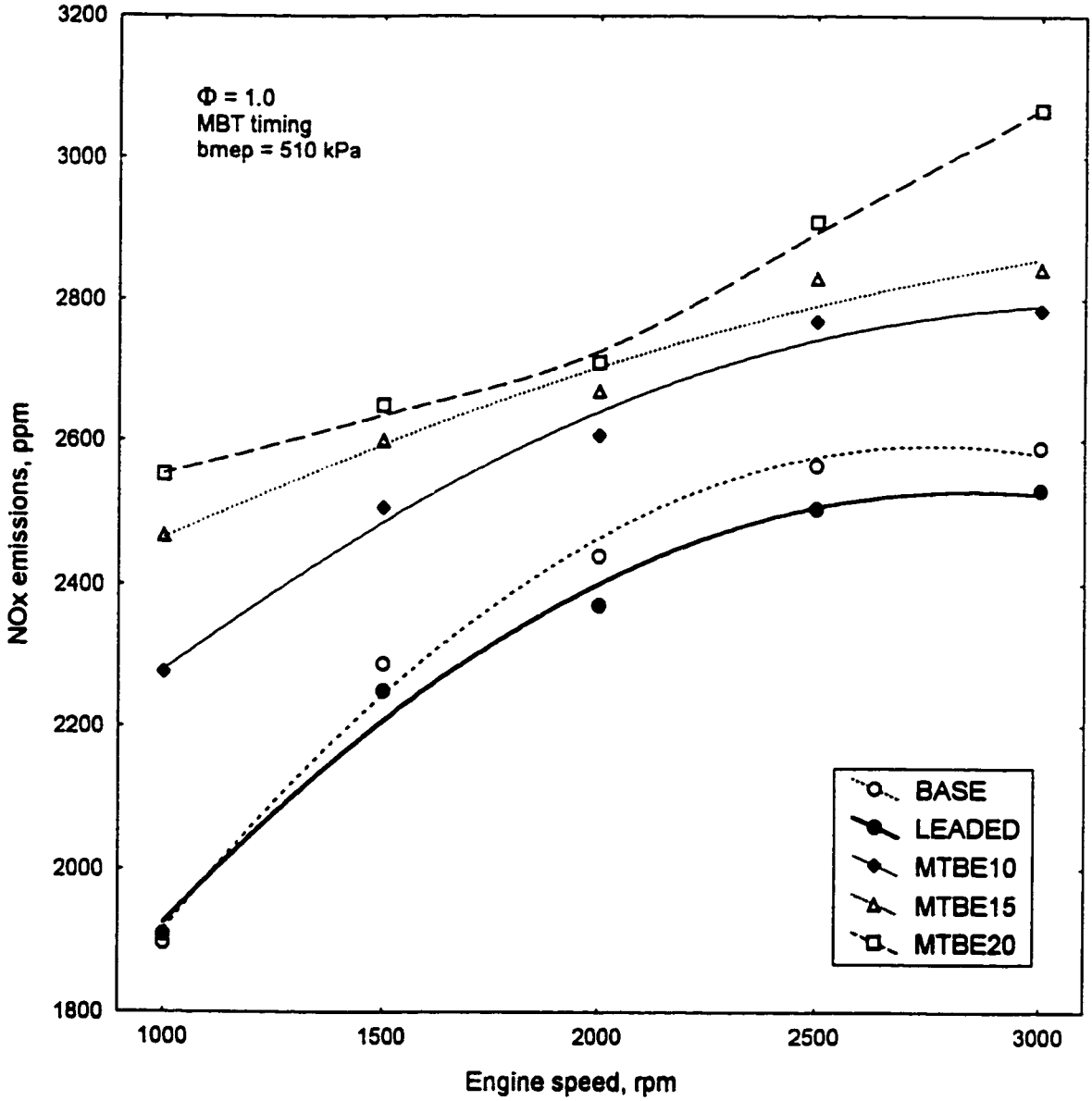

Figure 5-95: NOx emissions vs. speed at a constant load of 120 Nm (bmep = 510 kPa) for the MTBE blends.

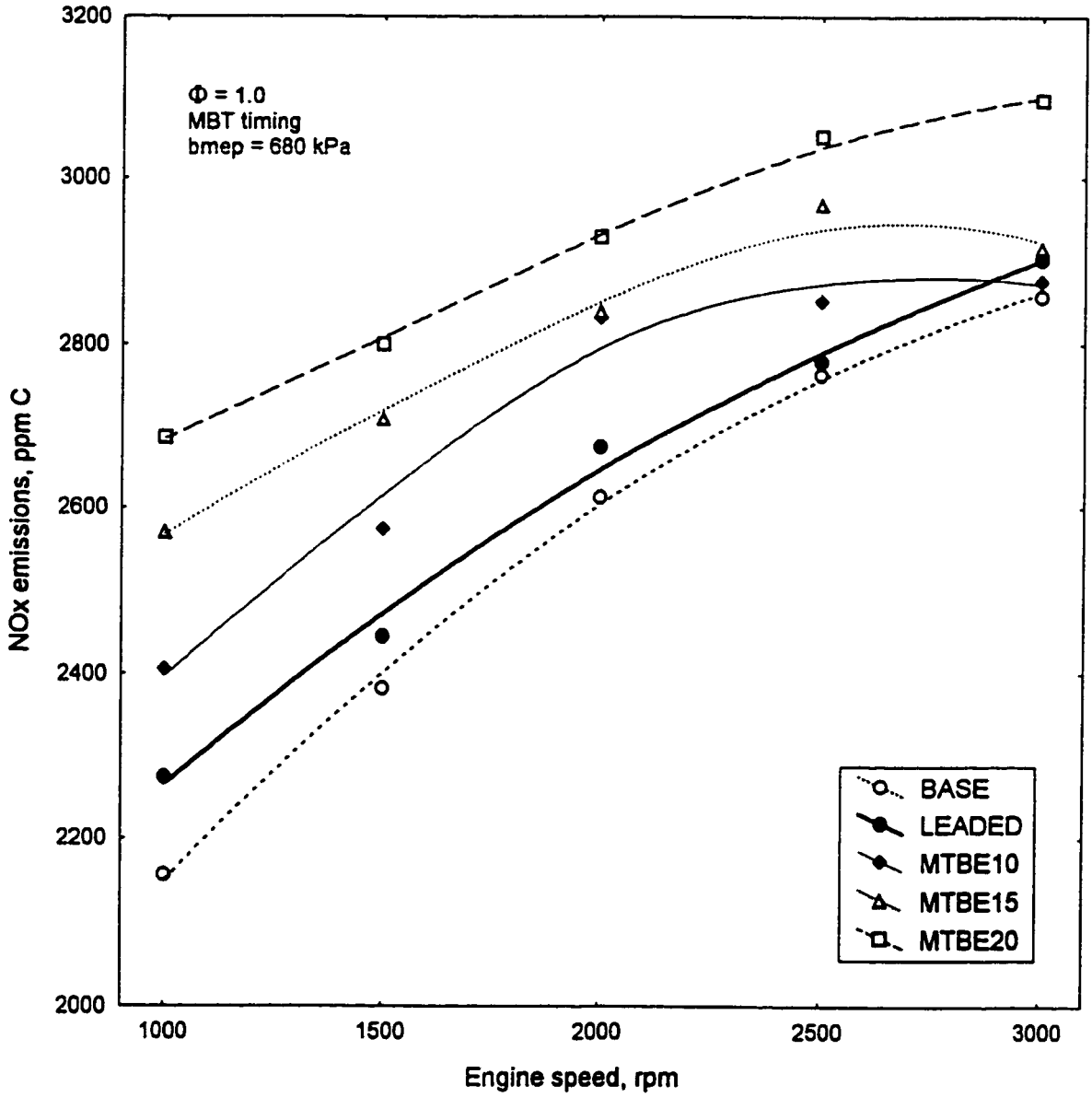

Figure 5-96: NOx emissions vs. speed at a constant load of 160 Nm (bmep = 680 kPa) for the MTBE blends.

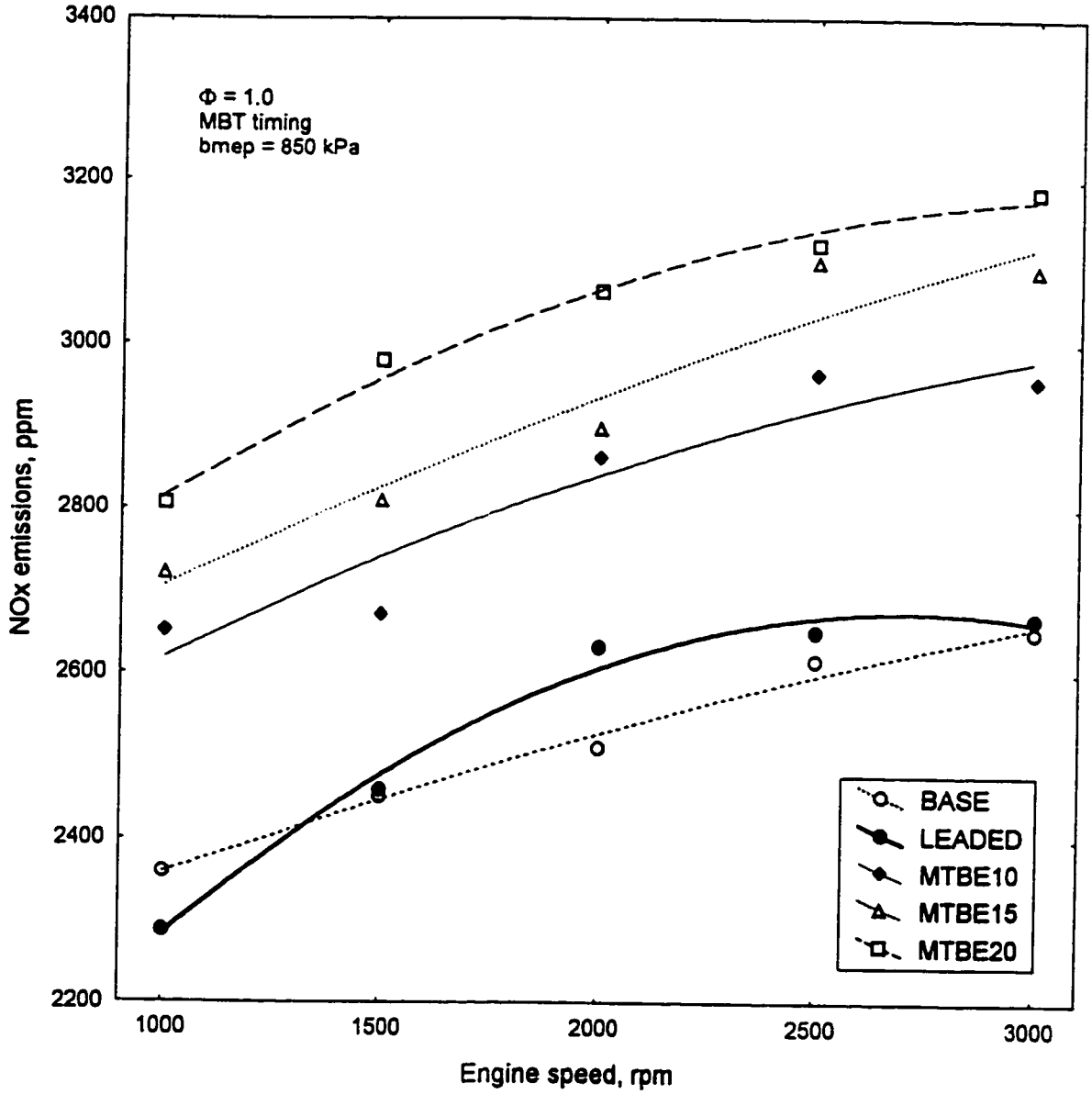

Figure 5-97: NOx emissions vs. speed at a constant load of 200 Nm (bmep = 850 kPa) for the MTBE blends.

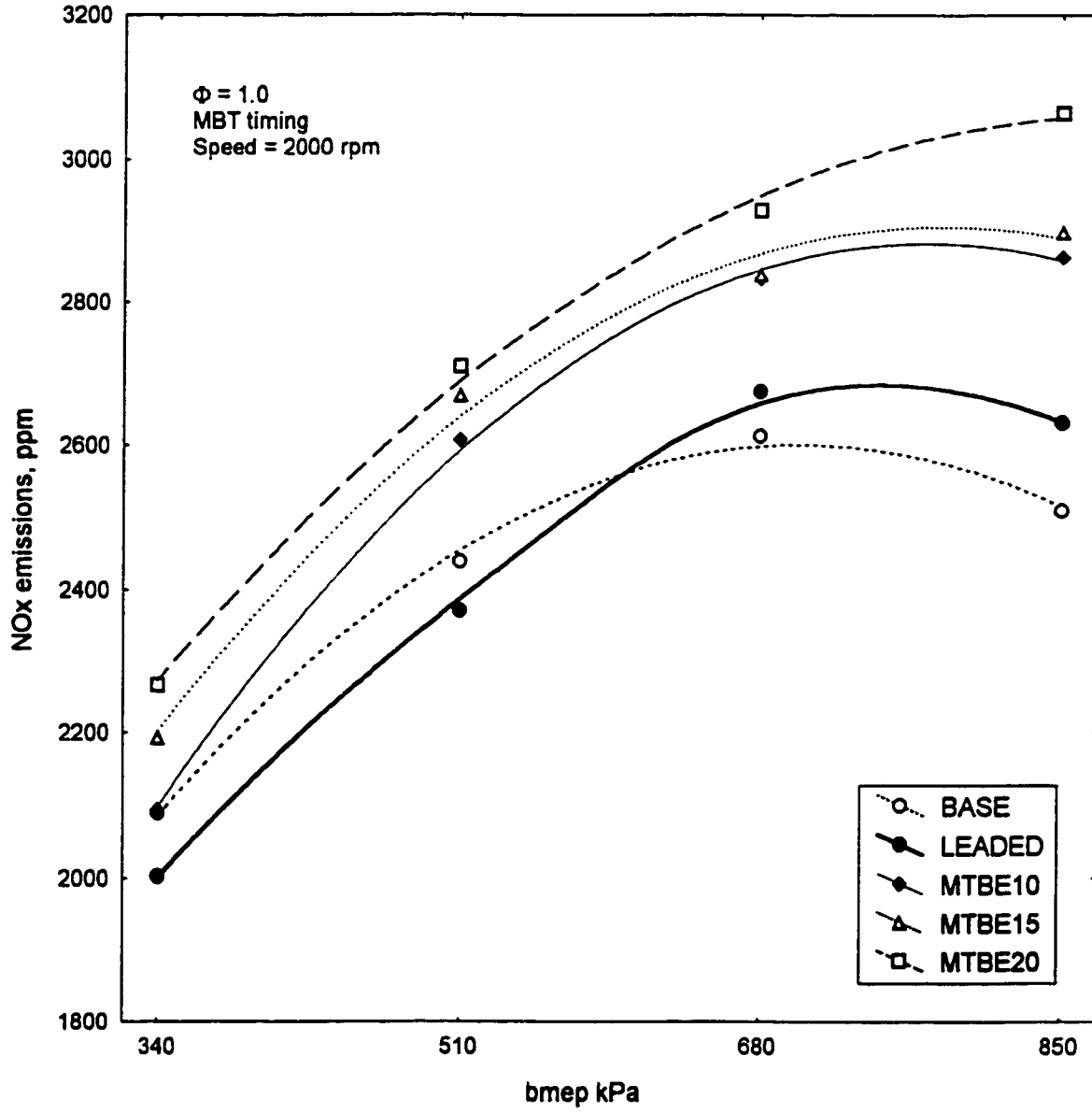

Figure 5-98: Effect of load on NOx emissions at a constant speed of 2000 rpm for the MTBE blends.

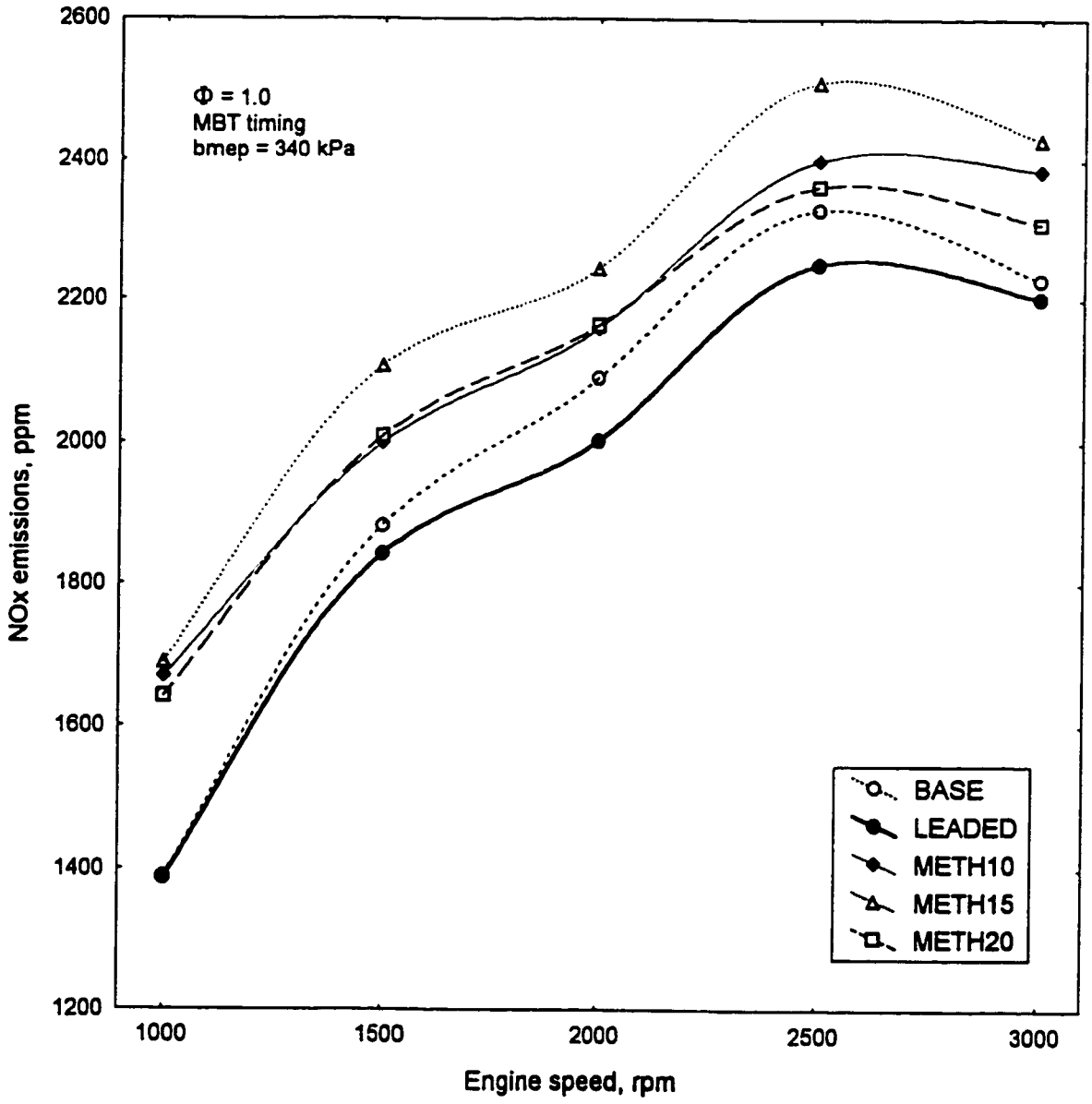

Figure 5-99: NOx emissions vs. speed at a constant load of 80 Nm (bmep = 340 kPa) for the methanol blends.

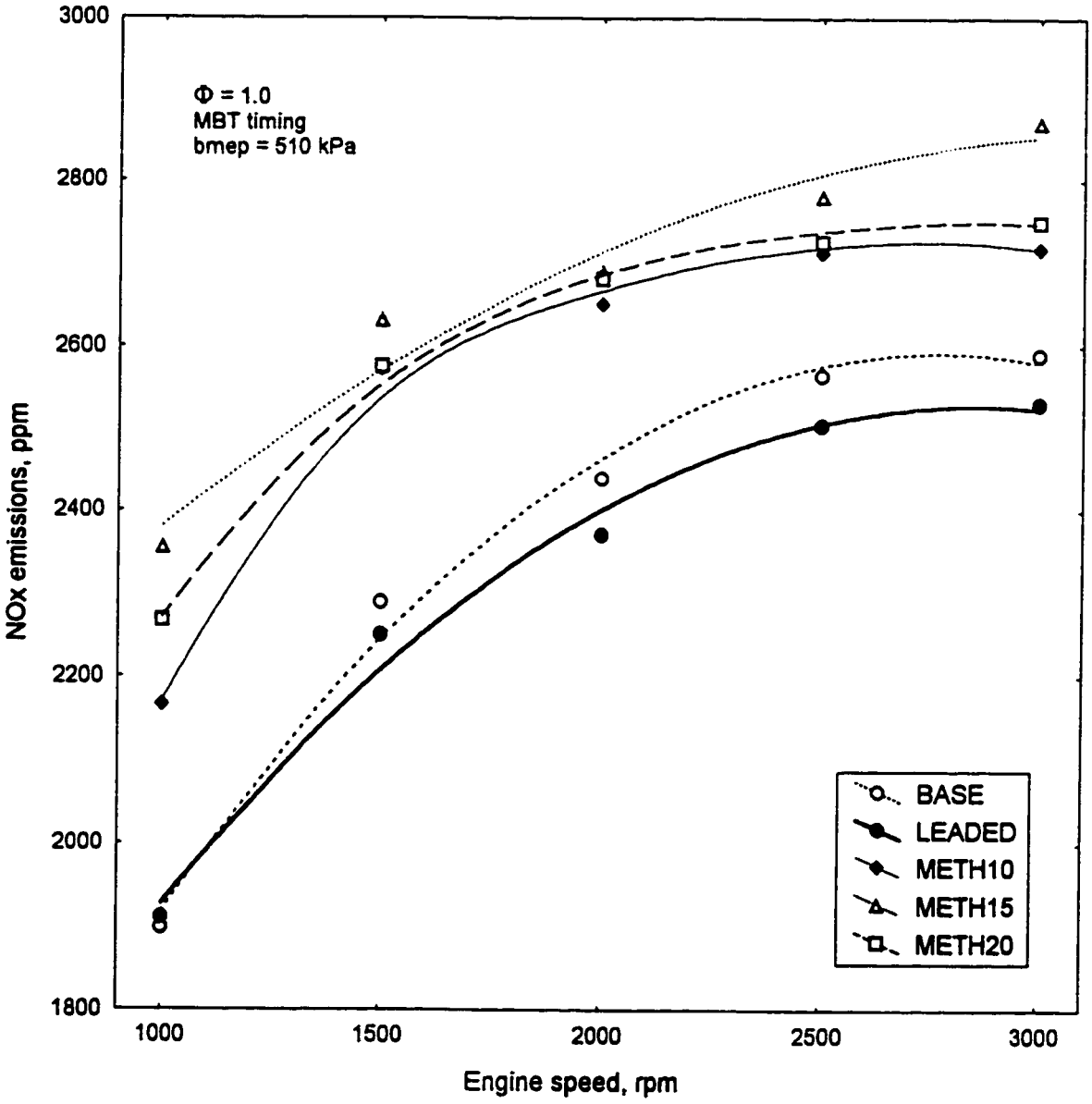

Figure 5-100: NOx emissions vs. speed at a constant load of 120 Nm (bmep = 510 kPa) for the methanol blends.

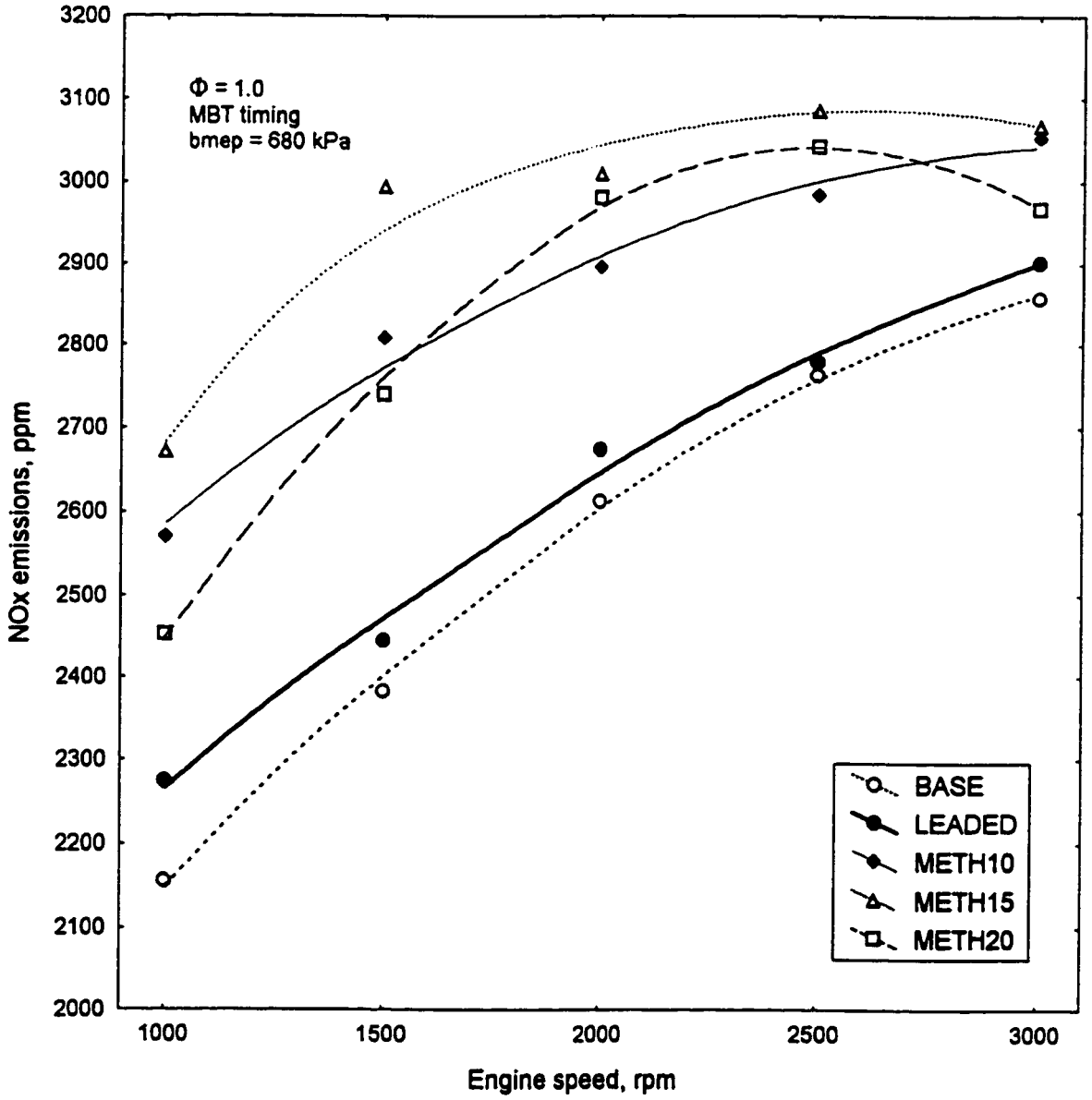

Figure 5-101: NOx emissions vs. speed at a constant load of 160 Nm (bmep = 680 kPa) for the methanol blends.

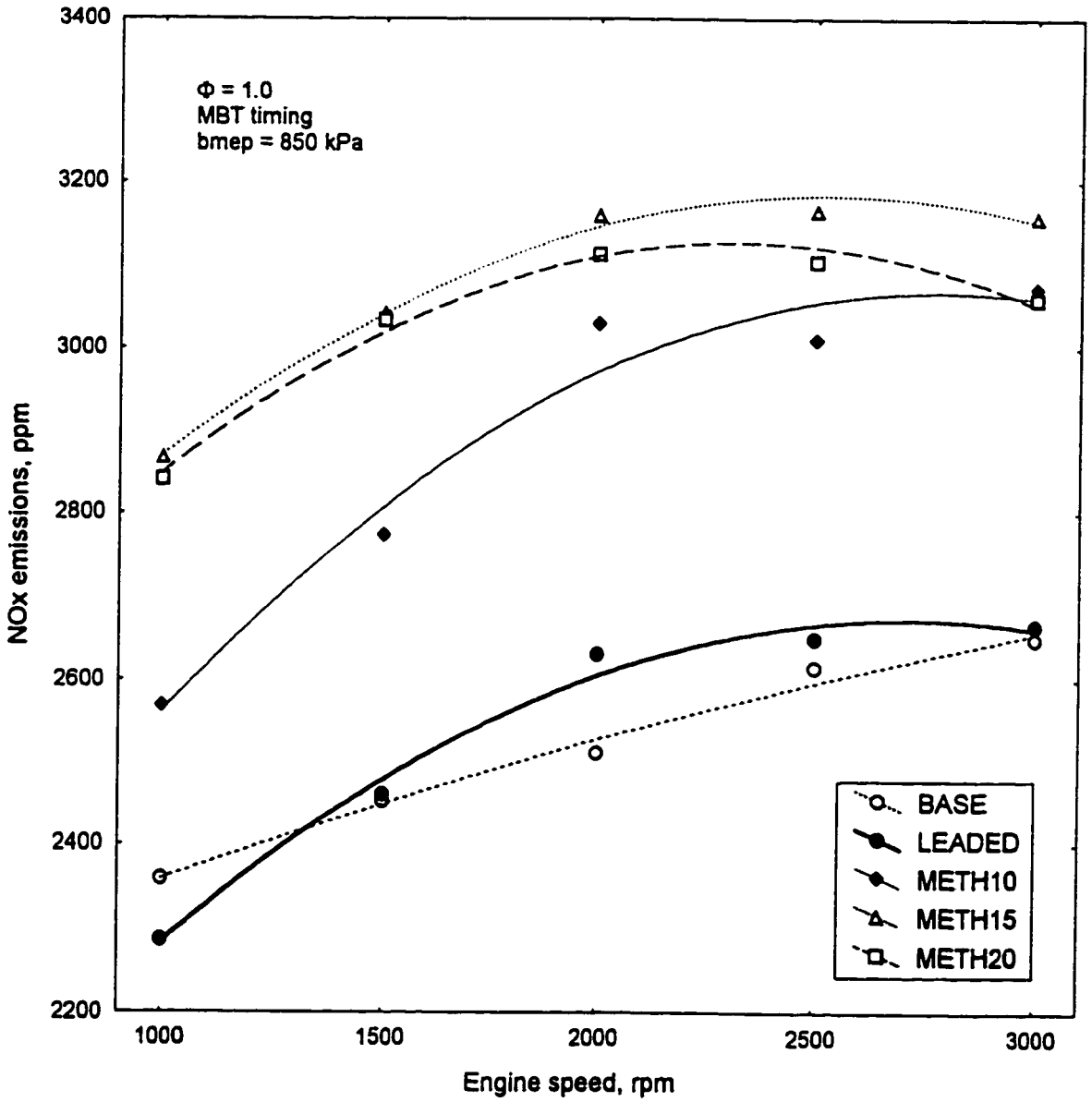

Figure 5-102: NOx emissions vs. speed at a constant load of 200 Nm (bmep = 850 kPa) for the methanol blends.

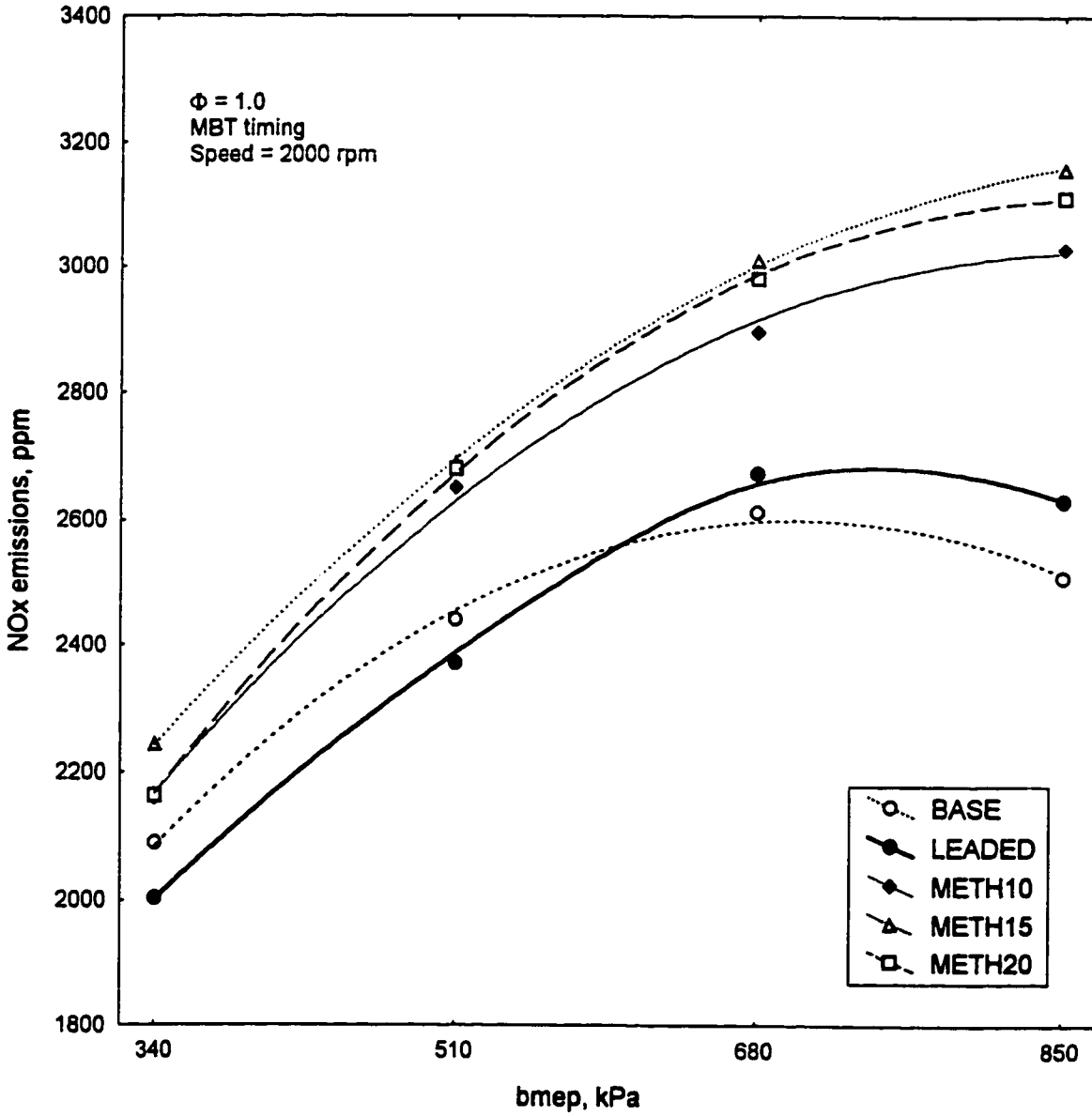

Figure 5-103: Effect of load on NOx emissions at a constant speed of 2000 rpm for the methanol blends.

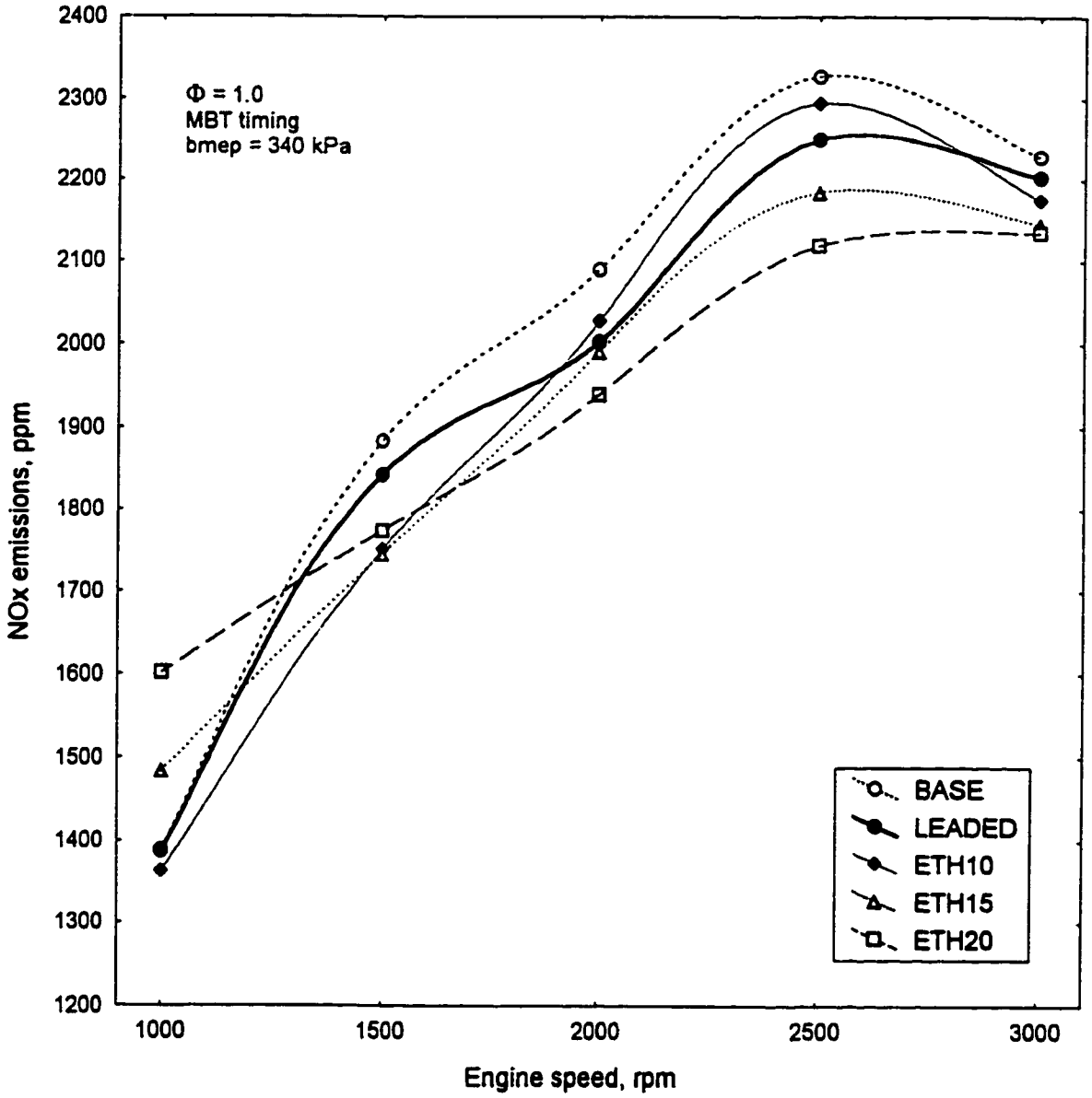

Figure 5-104: NOx emissions vs. speed at a constant load of 80 Nm (bmep = 340 kPa) for the ethanol blends.

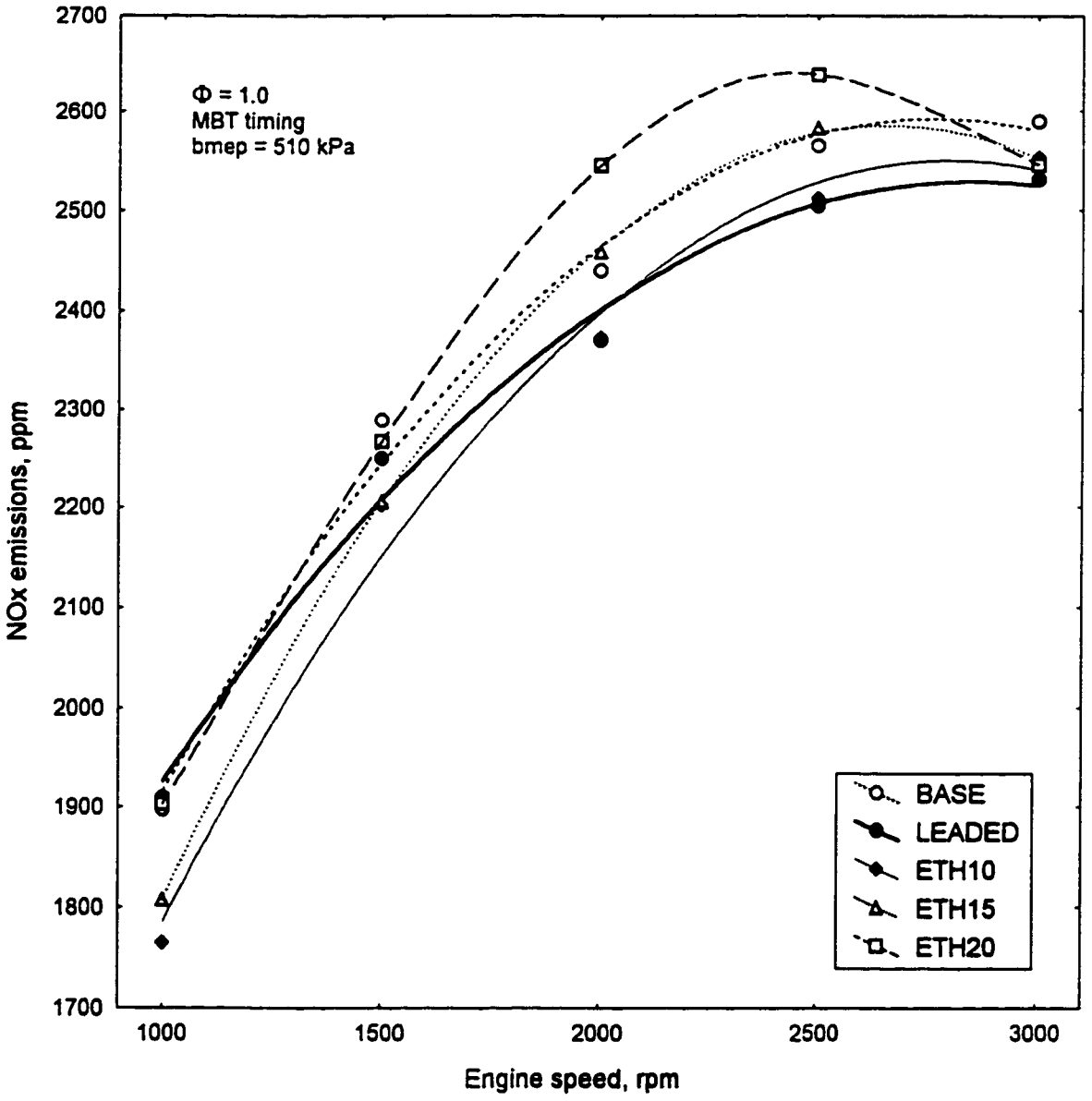

Figure 5-105: NOx emissions vs. speed at a constant load of 120 Nm (bmep = 510 kPa) for the ethanol blends.

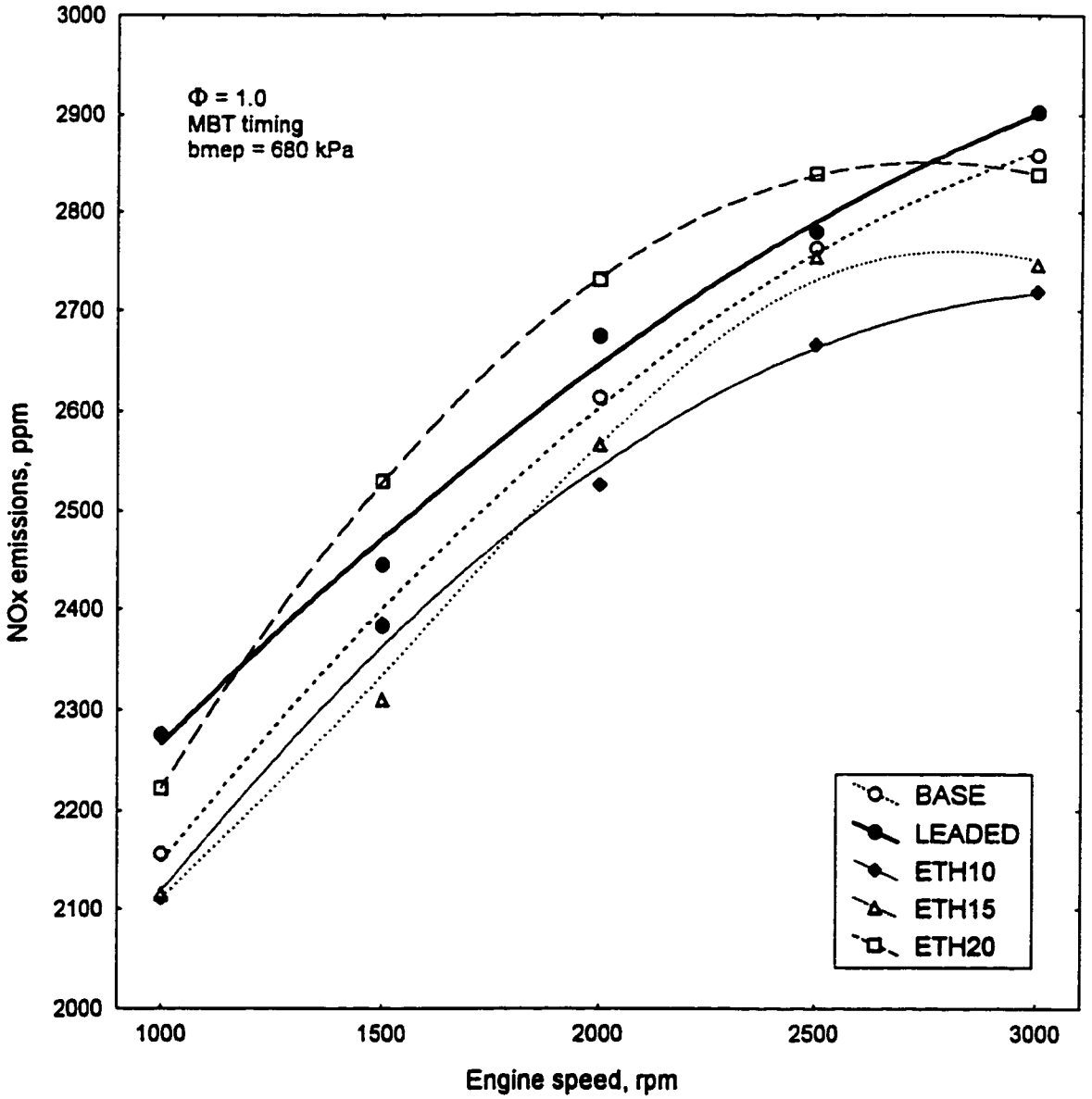

Figure 5-106: NOx emissions vs. speed at a constant load of 160 Nm (bmep = 680 kPa) for the ethanol blends.

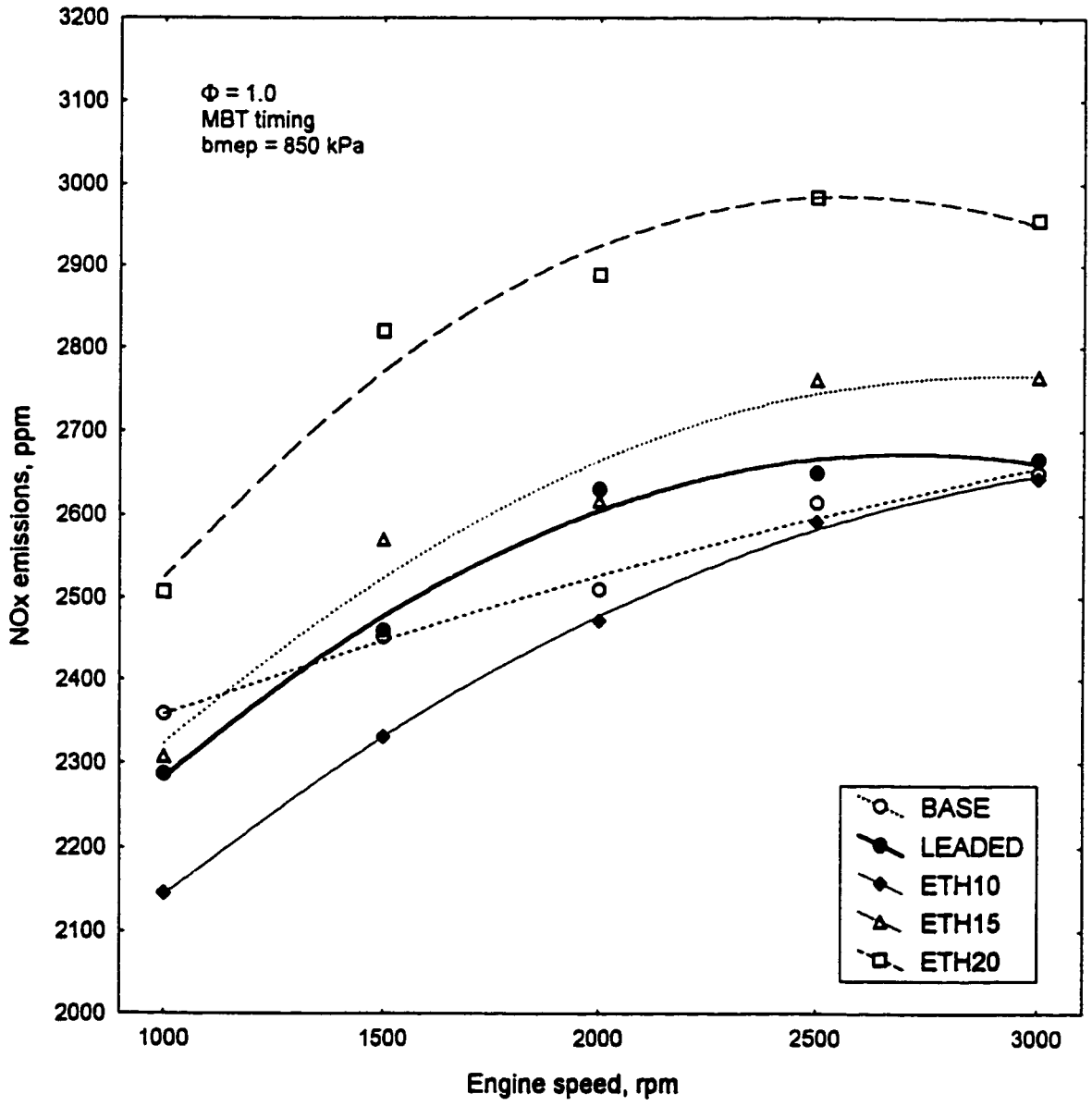

Figure 5-107: NOx emissions vs. speed at a constant load of 200 Nm (bmep = 850 kPa) for the ethanol blends.

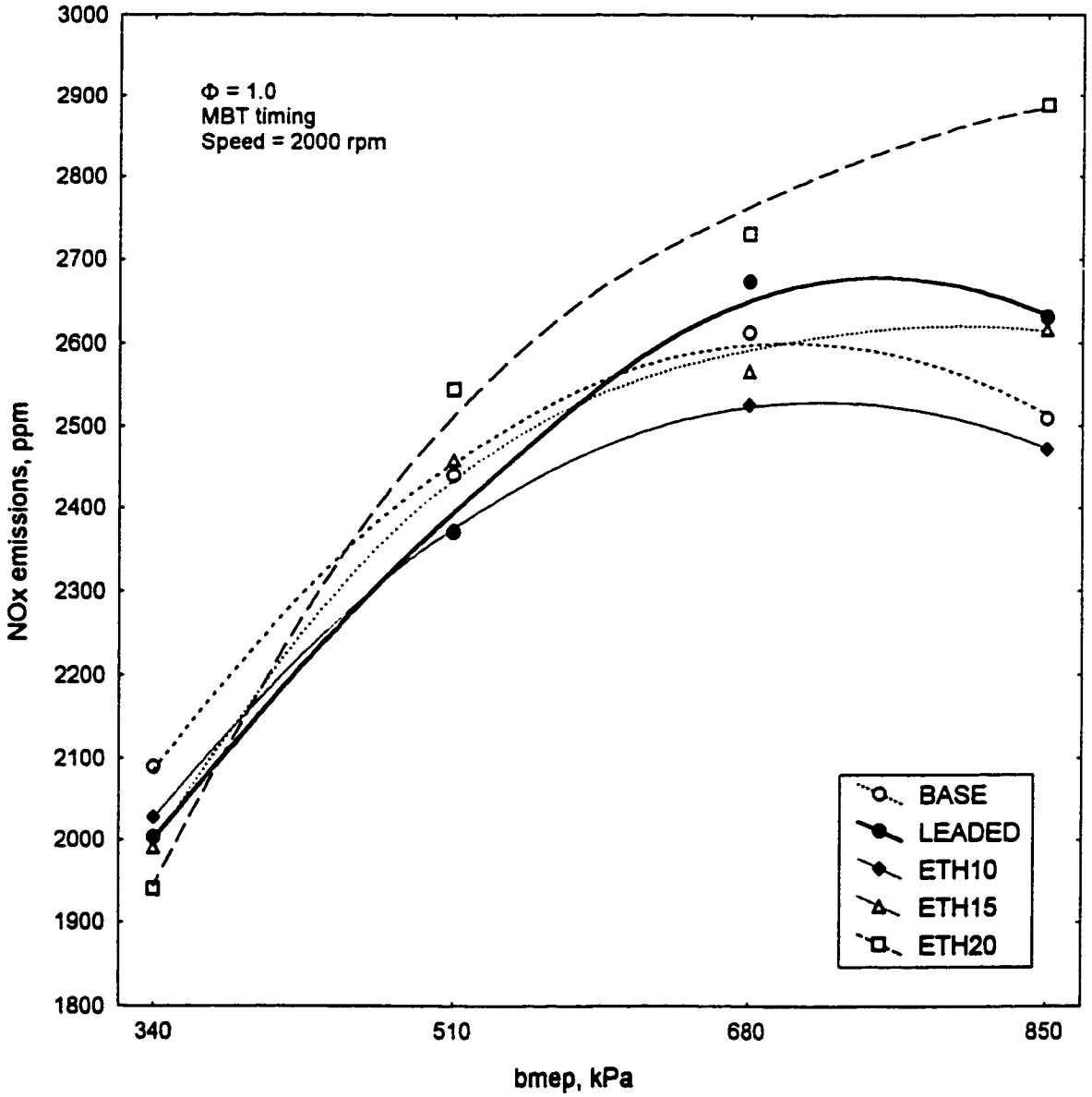

Figure 5-108: Effect of load on NOx emissions at a constant speed of 2000 rpm for the ethanol blends.

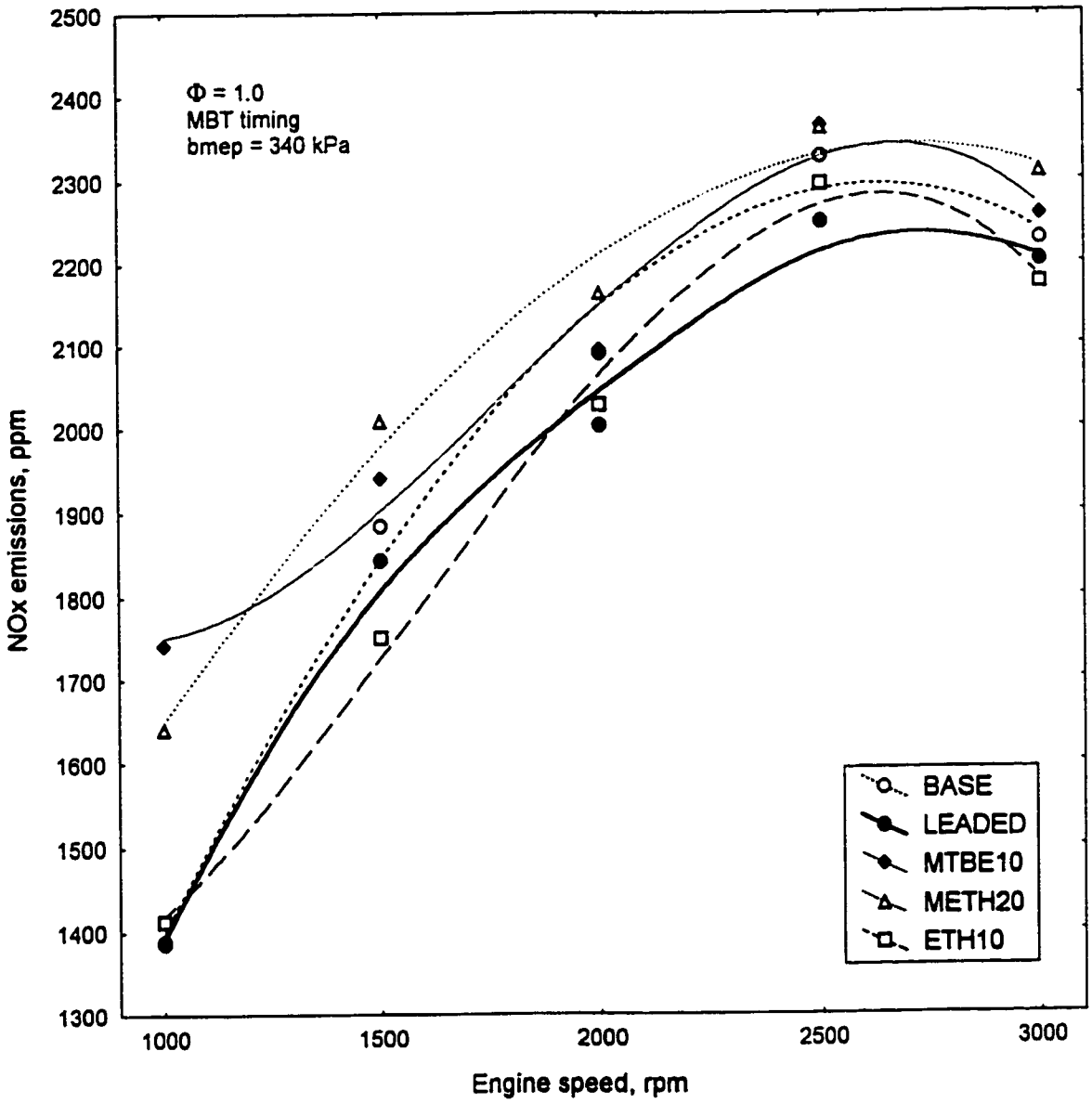

Figure 5-109: NOx emissions vs. speed at a constant load of 80 Nm (bmep = 340 kPa) for the least emitting blends compared to the base and leaded fuels.

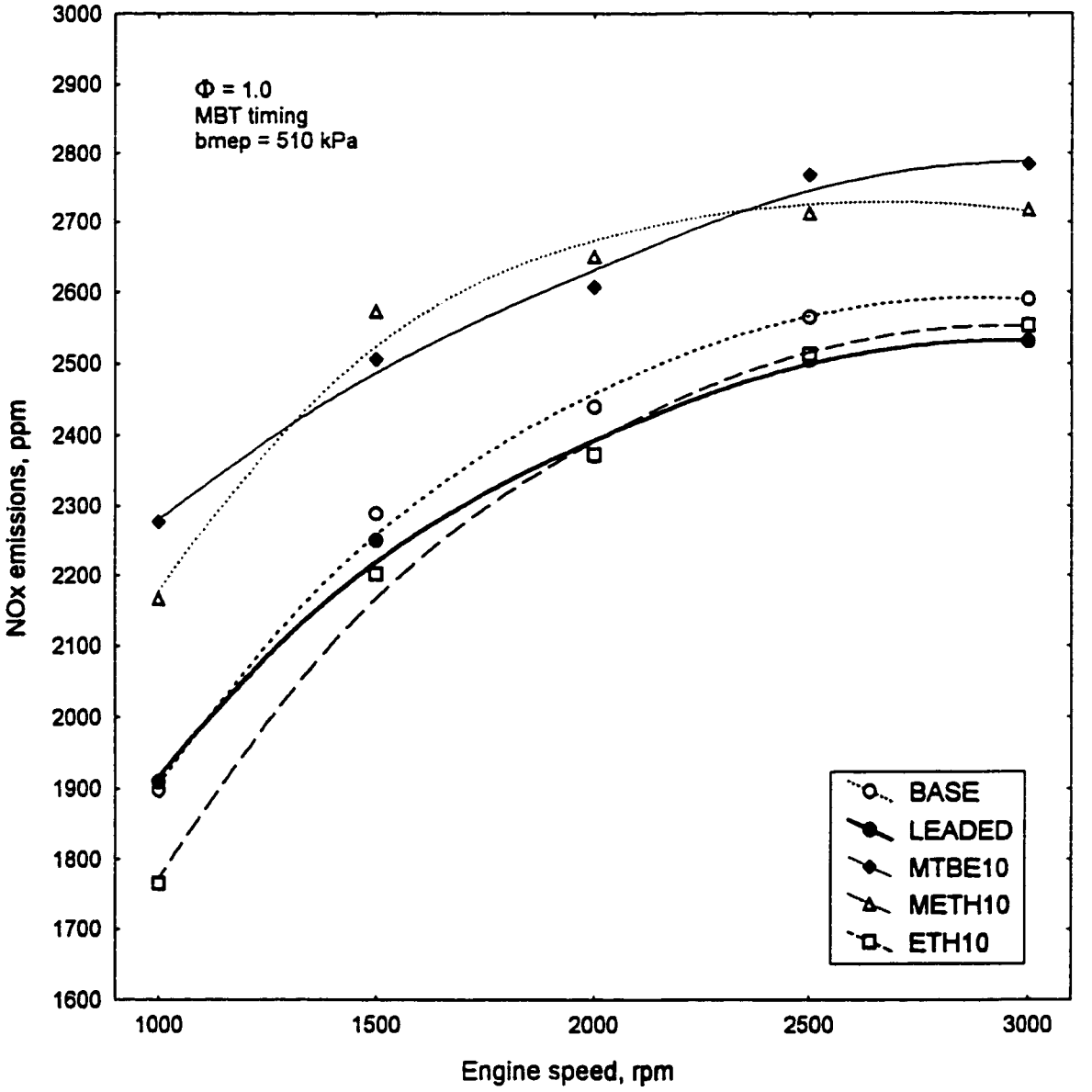

Figure 5-110: NOx emissions vs. speed at a constant load of 120 Nm (bmep = 510 kPa) for the least emitting blends compared to the base and leaded fuels.

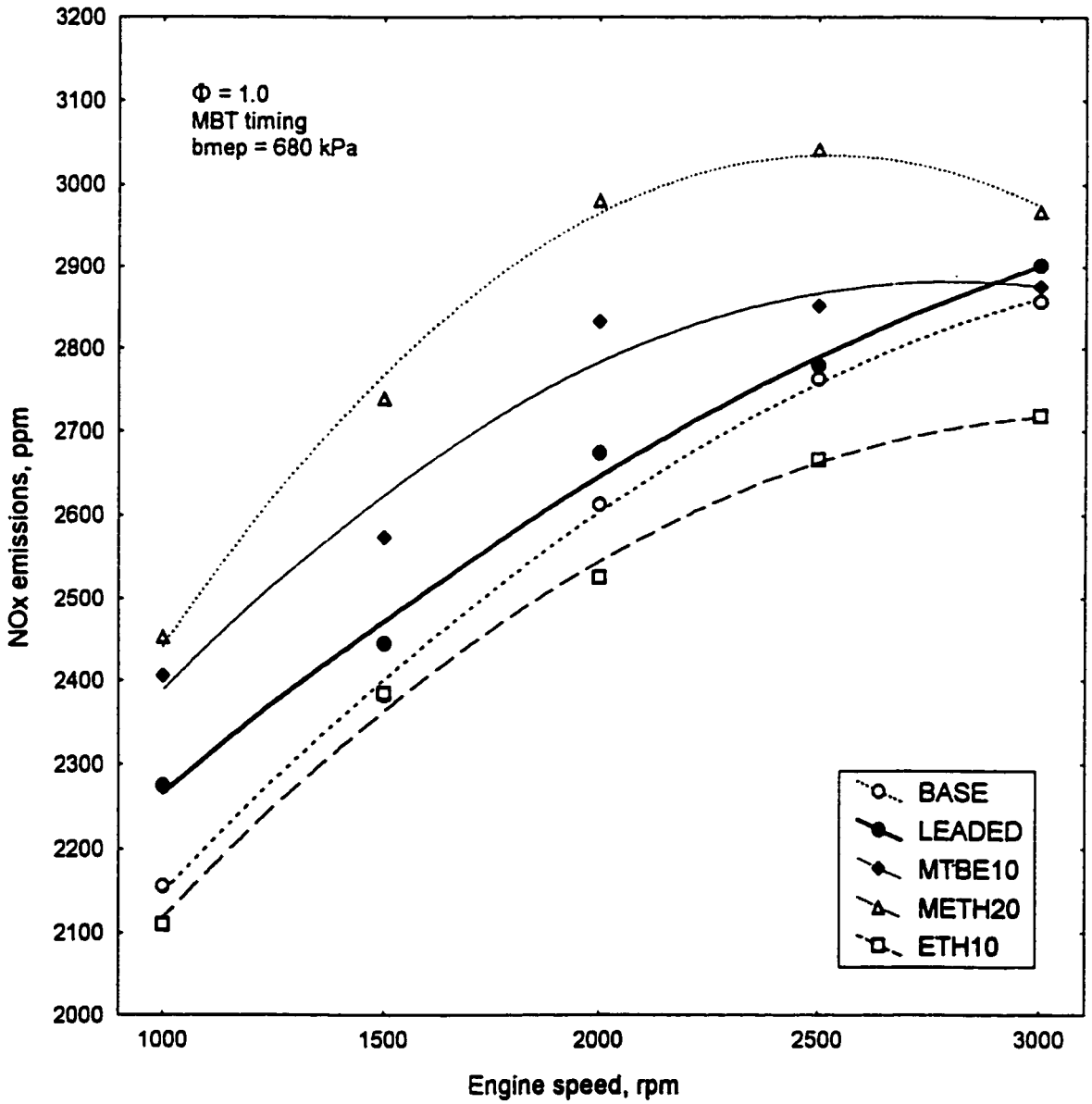

Figure 5-111: NOx emissions vs. speed at a constant load of 160 Nm (bmep = 680 kPa) for the least emitting blends compared to the base and leaded fuels.

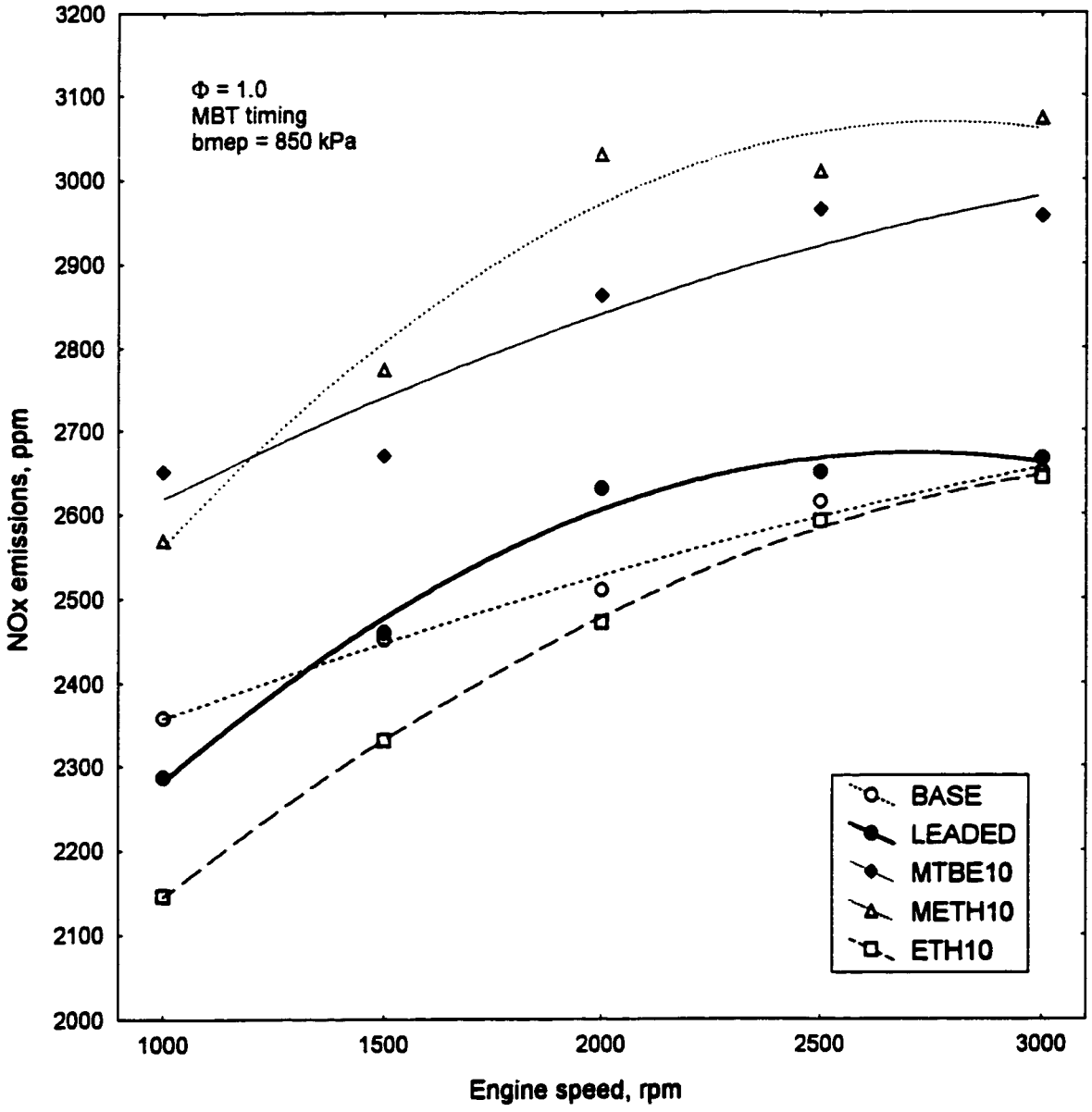

Figure 5-112: NOx emissions vs. speed at a constant load of 200 Nm (bmep = 850 kPa) for the least emitting blends compared to the base and leaded fuels.

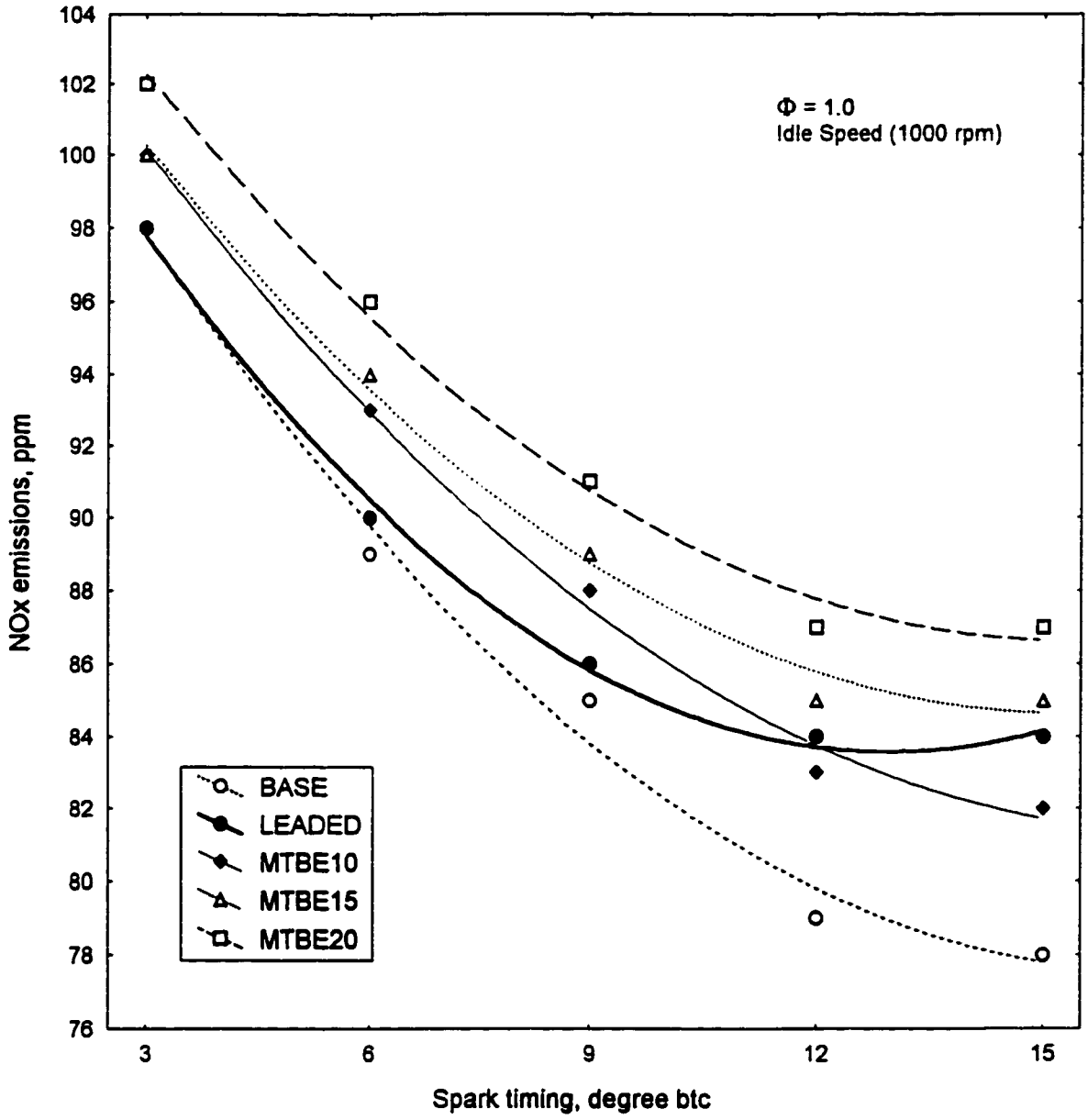

Figure 5-113: Effect of spark timing on NOx emissions at idle speed for the MTBE blends.

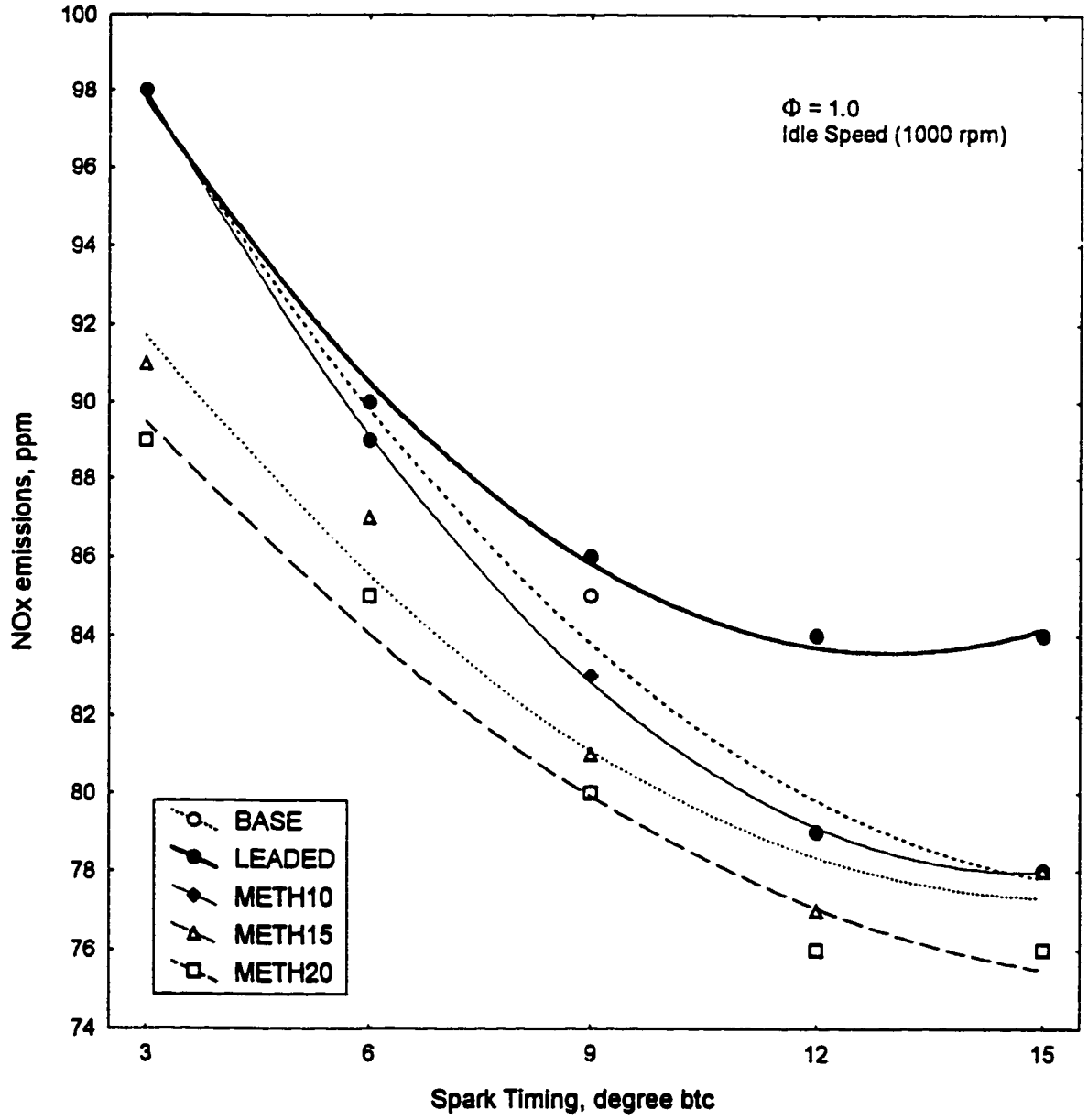

Figure 5-114: Effect of spark timing on NOx emissions at idle speed for the methanol blends.

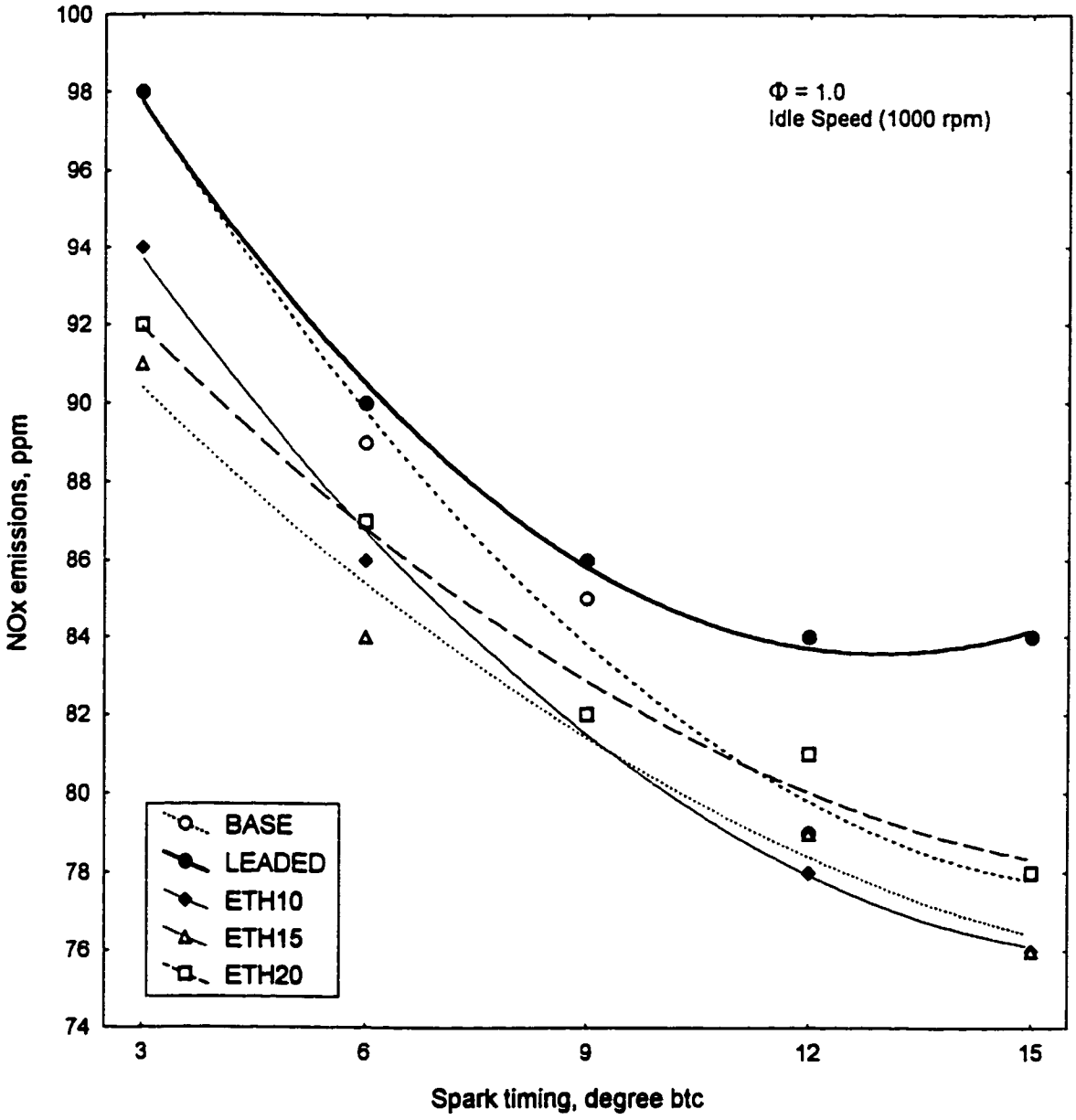

Figure 5-115: Effect of spark timing on NOx emissions at idle speed for the ethanol blends.

Chapter

6

Conclusions and Recommendations

6.1 Summary of Performance Results

6.1.1 Maximum Brake Mean Effective Pressure

The results from the variable-speed wide-open throttle (WOT) tests show that the leaded and oxygenated fuels performed better than the base fuel in terms of maximum brake mean effective pressure (bmep). The improvement in performance persists along the entire tested speed range. For leaded fuel, the average increase of maximum bmep with respect to the base fuel is about 4%. The improvement due to the addition of oxygenates varied with the oxygenate type and ratio in the blend but generally was less than that noticed with leaded fuel except in the case of the 20 vol.% methanol and 15 vol.% ethanol blends.

Figure 6.1 shows the effect of oxygenate type and ratio on the maximum bmep compared to the base (0 vol.% oxygenate) and leaded (straight-dashed line) fuels at a mid-range speed of 2500 rpm. The bmep was improved by addition of up to 15 vol.% of MTBE or ethanol but declined as the ratio increased. On the other hand, the bmep continued to improve as the methanol ratio in the blend increased. The improvement in bmep due to addition of 15 vol. % ethanol is equivalent to that of 20 vol.% methanol and was about 3.8% with respect to the base fuel. In the case of MTBE, the maximum improvement in bmep, which resulted from 15 vol.% MTBE, was about 1.8%.

For lower ratios of oxygenates (up to 15 vol.%), the ethanol blends performed better than methanol and MTBE blends. For higher ratios, however, the methanol was the best oxygenate in terms of maximum bmep of the engine.

6.1.2 Brake Thermal Efficiency at WOT

The results show that the oxygenated fuels resulted in higher brake thermal efficiency than the base fuel and than leaded fuel particularly at higher engine speeds. Although less than the oxygenated fuels, the thermal efficiency improved with leaded fuel but this improvement declined as the engine speed increased until eventually vanished at high speed of 3500 rpm.

Figure 6.2 shows the effect of oxygenate type and ratio on the WOT brake thermal efficiency compared to the base and leaded fuel at a mid-range speed of 2500 rpm. The thermal efficiency continued to increase as the oxygenate ratio increased for the three types. At this speed, the methanol blends were the best performers followed by MTBE then ethanol blends. Methanol blends resulted in maximum increase in thermal efficiency of bout 12.3% with respect to base fuel. The maximum increase in thermal efficiency is about 8.8% in the case of MTBE blends, and about 7.9% in the case of ethanol blends. Overall, the methanol blends were the best performers in terms of brake thermal efficiency at practical wide-open throttle conditions.

6.1.3 Brake Thermal Efficiency vs. Equivalence Ratio

The results from variable-equivalence ratio tests show that the differences between fuels are less significant at part-load than in the wide-open throttle condition. With all the tested fuels, the brake thermal efficiency improved significantly as the equivalence ratio is decreased (i.e. as the mixture is leaned out).

Figure 6.3 shows the effect of oxygenate type and ratio on the brake thermal efficiency compared to the base and leaded fuels at a stoichiometric mixture (Φ=1.0), constant load of 680 kPa, and constant speed of 2000 rpm. At these conditions, the best oxygenate in terms of brake thermal efficiency was ethanol for low oxygenate ratio (up to 10 vol.%) and methanol for higher ratio (15 and 20 vol.%). The maximum increase in thermal efficiency with respect to the base fuel was about 4.7% in the case of methanol,

about 4.0% in the case of MTBE, about 2.3% in the case of ethanol, and about 3.3% in the case of leaded fuel.

6.2 Summary of Emissions Results

6.2.1 CO Emissions

Except in the case of lean mixtures, the results generally indicate a noticeable reduction in CO emissions due to the addition of the oxygeantes over the whole range of spark timing, speed and load. The CO concentration continued to decrease as the oxygenate ratio in the blend increases. With lean mixtures, however, no differences are noticed between CO emissions results of the tested fuels.

At lower loads (340-510 kPa), the MTBE20 blend was the least CO emitting in general with a maximum reduction with respect to the base fuel of about 24% at bmep = 340 kPa and 18% at bmep = 510 kPa. At higher loads (680-850 kPa), the least CO emitting blend was the ETH20 with a maximum reduction of about 12.6% at bmep = 680 kPa and 11.8% at bmep = 850 kPa. The leaded fuel generally resulted in higher CO emissions than the oxygenated blends. Figures 6-4 and 6-5 show the effect of the oxygenates on CO emissions at mid-range speed (2000 rpm) and two different loads (340 and 680 kPa).

At idle speed, a maximum reduction of about 35% in CO with respect to the base fuel is achieved by MTBE20. The results for METH20 came close to those of MTBE20, while ETH20 resulted in less significant reduction in CO particularly at advanced spark timing. The CO results for the leaded fuel are lower than those of the base fuel at late spark timing (close to the top center), but the results for the two fuels become closer to each other as the spark timing is advanced. Figures 6-6 and 6-7 show the effect of the oxygenates on CO emissions at idle speed with two different spark timings (3 and 9 degrees btc).

6.2.2 HC Emissions

The results for the HC emissions generally indicate that the oxygenated blends decreased HC emissions with respect to the base and leaded fuels except at very high loads and at very advanced spark timing. The results for the leaded fuel are comparable to those of the base with slight increase or decrease in some cases.

For loads up to 680 kPa, the least HC emitting blend is the MTBE20 with a maximum reduction of about 15.5% with respect to the base fuel. The difference between the blends becomes more evident as the load is increased. For the highest load, the lowest HC concentration resulted from the METH20 at lower speeds (a reduction of about 28.6%) and from the MTBE20 at higher speeds (a reduction of about 12.5%). Figures 6-8 and 6-9 show the effect of the oxygenates on HC emissions at mid-range speed (2000 rpm) and two different loads (340 and 680 kPa).

At idle speed, the METH20 had the lowest concentration of HC emissions among the blends with a maximum reduction of about 31% with respect to the base fuel (at spark timing = 9 degrees btc). The leaded fuel resulted in higher HC emissions than the base fuel except at very late spark timing (close to the top center) where a lower concentration is noticed. Figures 6-10 and 6-11 show the effect of the oxygenates on HC emissions at idle speed with two different spark timings (3 and 9 degrees btc).

6.2.3 NOx Emissions

In general, the oxygenated blends resulted in higher NOx with rich and stoichiometric mixtures but lower NOx with lean mixtures. The 10 vol.% ethanol is an exception with a consistent decrease at most operating conditions. The addition of 10 vol.% of MTBE or methanol caused the NOx to increase when compared to the base fuel. But while increasing MTBE resulted in more NOx emissions, increasing the methanol ratio in the blend caused the NOx to decrease. On the contrary, the addition of 10 vol.% ethanol resulted in less NOx, but increasing the ethanol caused the NOx to either increase or decrease depending on the test conditions. These complicated effects may explain some

of the conflict in the published results regarding the effect of oxygenates on NOx emissions.

The results show that the least NOx emitting blend, overall, is the ETH10. This blend caused a reduction in the NOx emissions with respect to the base fuel at most tested operating conditions. The other blends resulted generally in higher NOx emissions with respect to the base fuel. The results show that the leaded fuel in most cases resulted in higher NOx emissions than the base and ethanol blends and lower than the methanol and MTBE blends.

Figures 6-12 and 6-13 show the effect of the oxygenates on NOx emissions at mid-range speed (2000 rpm) and two different loads (340 and 680 kPa). Up to 15 vol.% of oxygenate in the blend, the methanol blends were the most NOx emitters at all operating conditions. For 20 vol.% oxygenated blends, the MTBE20 resulted in more NOx emissions than any other blend.

At idle speed the addition of oxygenates has no significant effect on NOx concentration which is already very small.

6.3 Conclusions

The addition of MTBE, methanol, or ethanol to the base gasoline affected the engine performance positively. The addition of these oxygenates improved the thermal efficiency and increased the maximum output of the engine. The best engine performance was noticed with the high-ratio methanol blends followed by ethanol then MTBE blends. When compare to the leaded fuel, the oxygenated blends resulted in higher thermal efficiency but, in general, lower maximum engine output except in the case of the 20 vol.% methanol and 15 vol.% ethanol blends.

The results showed that the addition of the oxygenates had different effects on the major exhaust emissions. In general, the addition of oxygenates decreased the carbon monoxide emissions (CO). It also decreased the hydrocarbon emissions (HC) at most

operating conditions. On the other hand, the addition of oxygenates generally increased the emissions of nitrogen oxides (NOx) except for lower-ratio ethanol blends.

The effects of oxygenates on the exhaust emissions varied with different engine operating conditions These variations with operating conditions may explain some of the conflict in published results which are based on very limited test conditions.

The emissions results indicate that the 20 vol.% ethanol blend was the least CO emitter at most conditions except at lower loads and idle where the 20 vol.% MTBE blend was the least emitter. The MTBE blends were the least HC emitters at most practical operating conditions. The results indicate, also, that the ethanol blends are the least NOx emitters among the tested blends. While the ethanol blends reduce or have no significant effect on NOx emissions, the methanol and MTBE blends increase these emissions.

The adoption of an alternative fuel depends on several factors other than the effects on engine performance and emissions. Factors such as the fuel economics, the material compatibility, and the handling issues are vital in deciding the practicality of a fuel. But even if these factors are disregarded and the selection of the best alternative fuel is based only on the tested quantities related to the engine performance and emissions, the selection will depend on the weight assigned to each quantity. Performance quantities, such as engine output and thermal efficiency, may be emphasized in some applications while the emissions quantities may be emphasized in others. It is save in most of the time to choose the fuel that represent the best compromise between the different tested quantities.

Overall, the methanol blends performed better than the other blends in terms of engine output and thermal efficiency. The MTBE blends were the best performers in terms of HC emissions while the ethanol blends were the best performers in terms of CO and NOx emissions. However, in the light of the results of the current study, it seems that the 15 vol.% ethanol (ETH15) blend is the best compromise between engine

performance and exhaust emissions. The ETH15 blend gives a significant improvement in engine maximum output and thermal efficiency, significantly reduces CO emissions, moderately reduces HC emissions, and generally has no effect on NOx emissions.

6.4 Recommendations

The results of the steady-state static tests conducted in the current study suggest the technical feasibility of replacing lead additives with oxygenated blends. However, in order to have the complete picture, further studies are needed to expose the effects of oxygenated blends in the transient conditions. Transient conditions such as the engine cold-starting, warm-up, and acceleration represent a substantial part of engine operation. Studying the transient effects will help in the assessment of the practicality of the oxygenated blends.

Although the effects of influential engine operating variables were well investigated in the current study, the effects of engine design parameters have not been a part of the study. The study was conducted on only one fuel-injected six-cylinder engine. In order to generalize the results, the study must be extended to several engines with several design configurations.

The current study investigates the effects of oxygenates on the engine performance and emissions but does not take in consideration issues such as the possible incompatibility with fuel system materials, the toxicity, and water-intolerance. These issues are important factors in deciding the practicality of a certain oxygenate. Therefore, it is imperative to extensively explore these issues before deciding what oxygenate to use as a replacement of the lead additives.

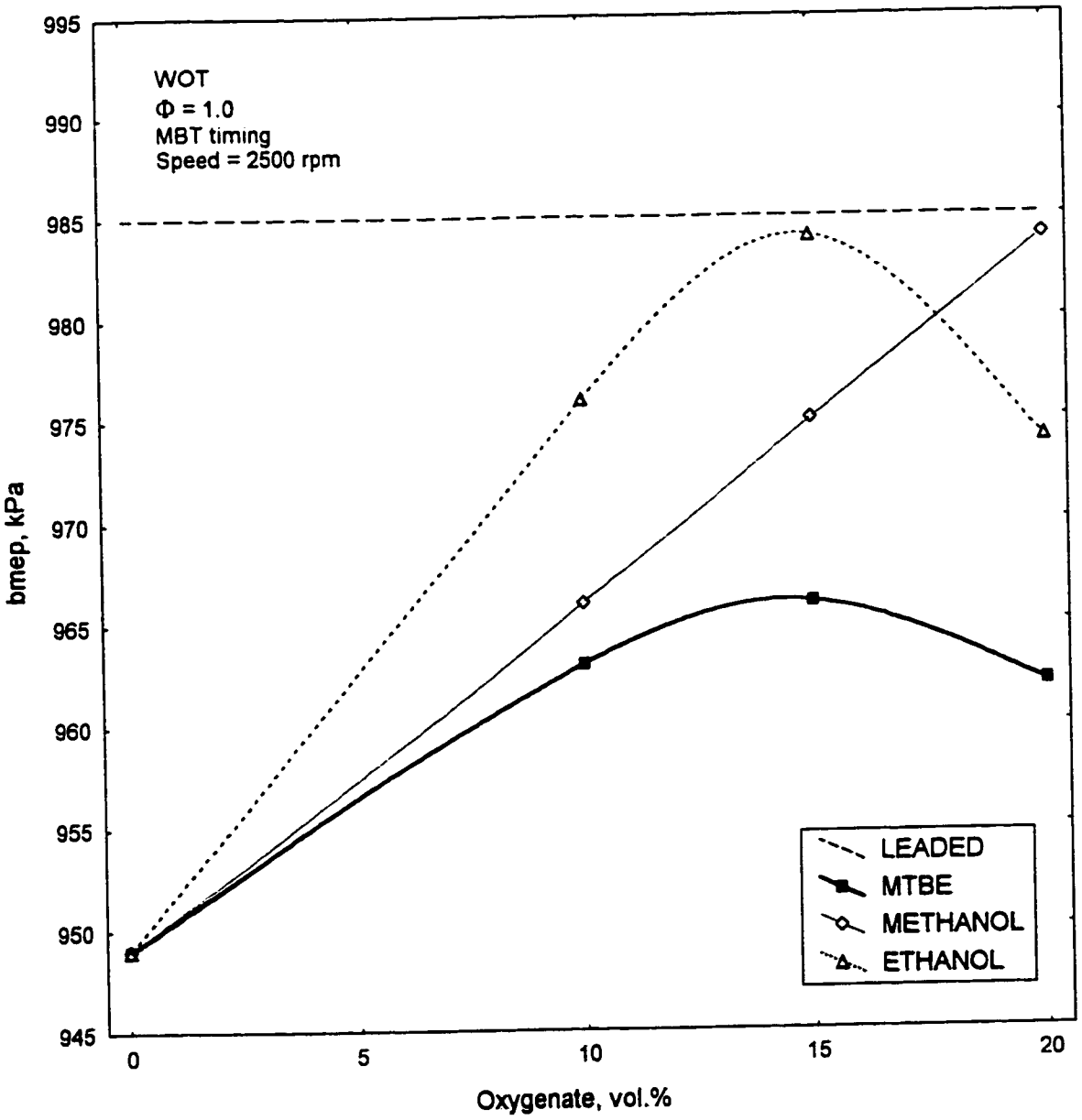

Figure 6-1: Effect of the oxygenates on brake mean effective pressure (bmep) at 2500 rpm and wide-open throttle.

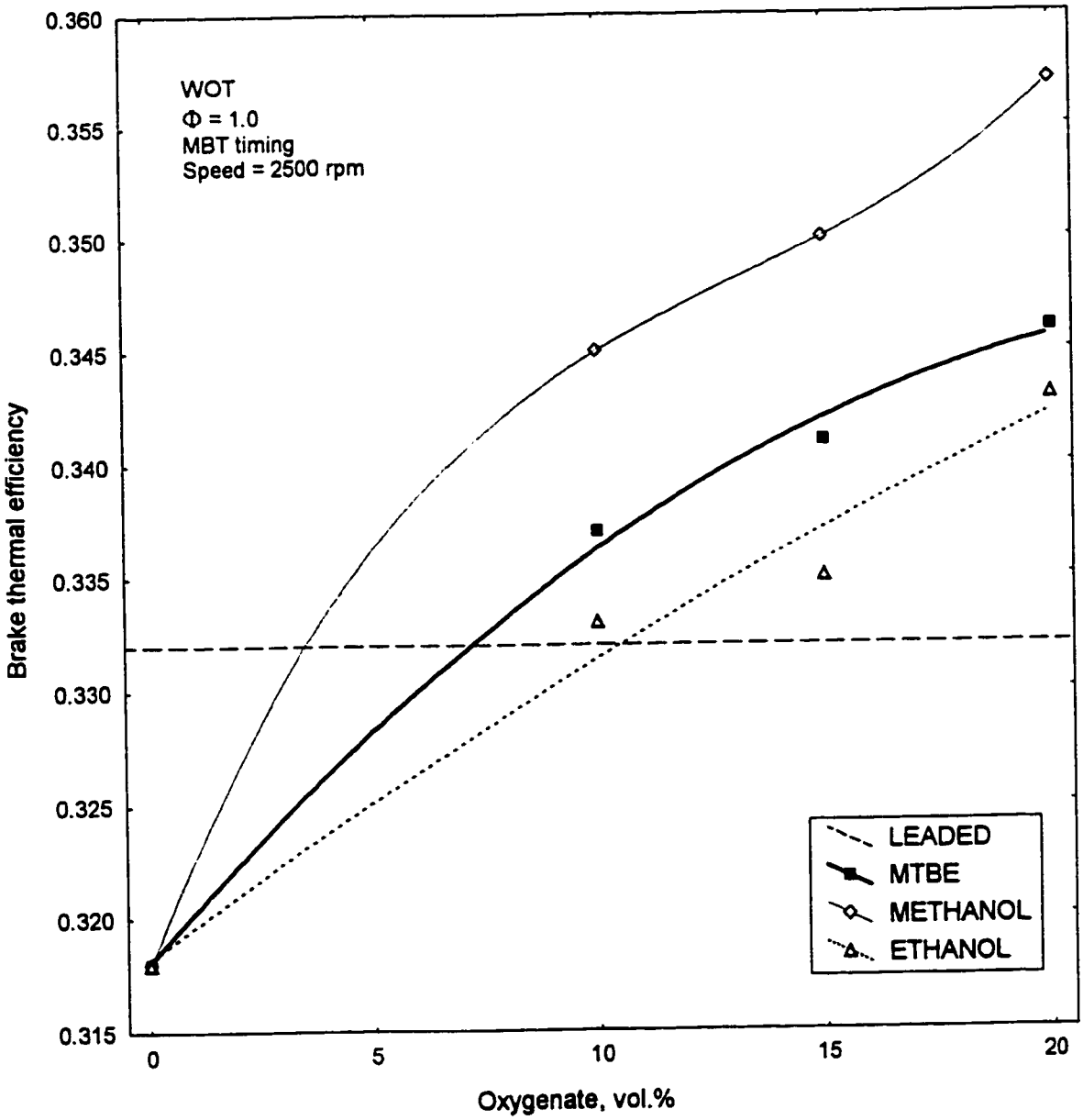

Figure 6-2: Effect of the oxygenates on brake thermal efficiency at 2500 rpm and wide-open throttle.

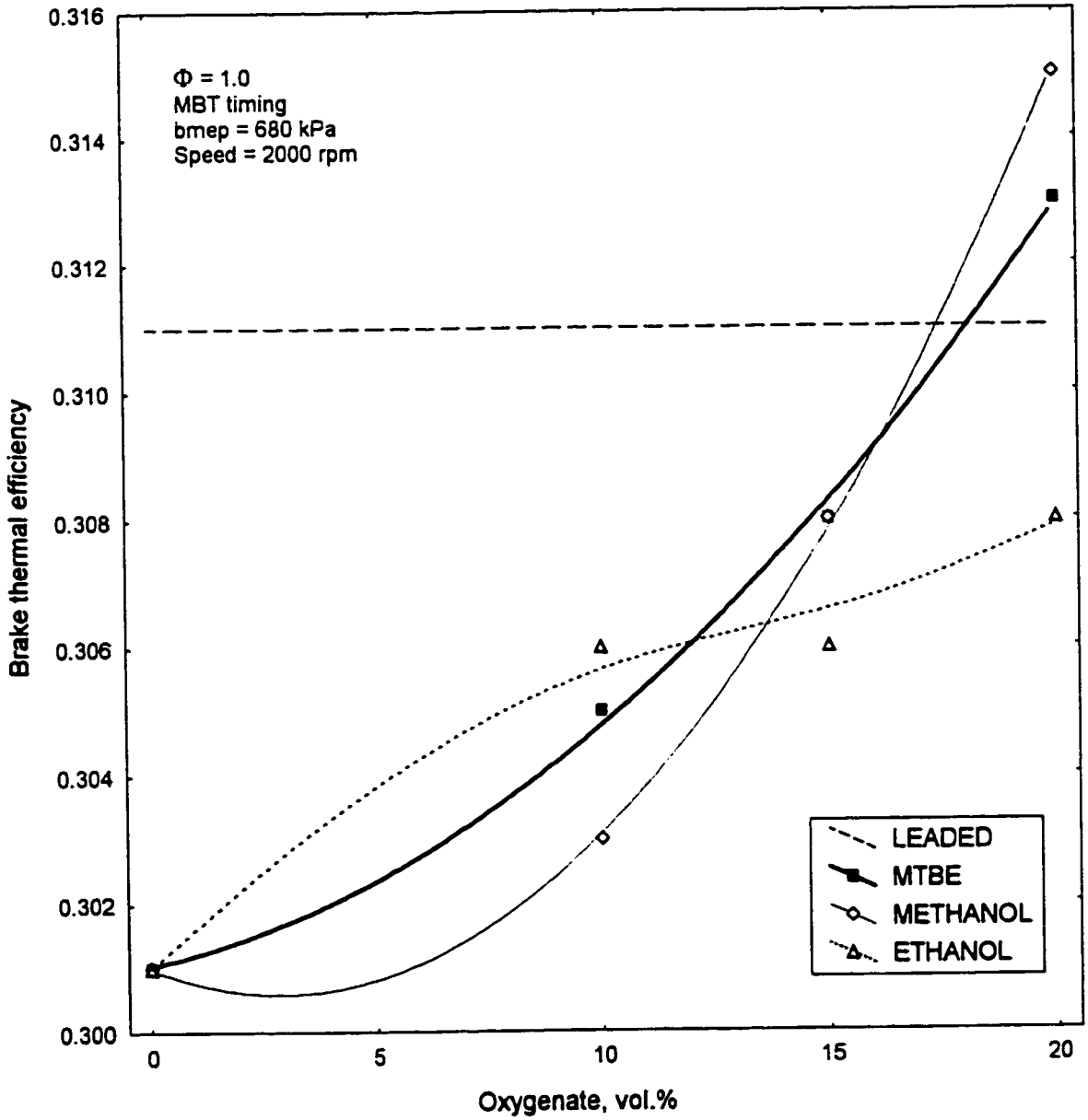

Figure 6-3: Effect of the oxygenates on brake thermal efficiency for stoichiometric mixtures (Φ=1.0) at 2000 rpm.

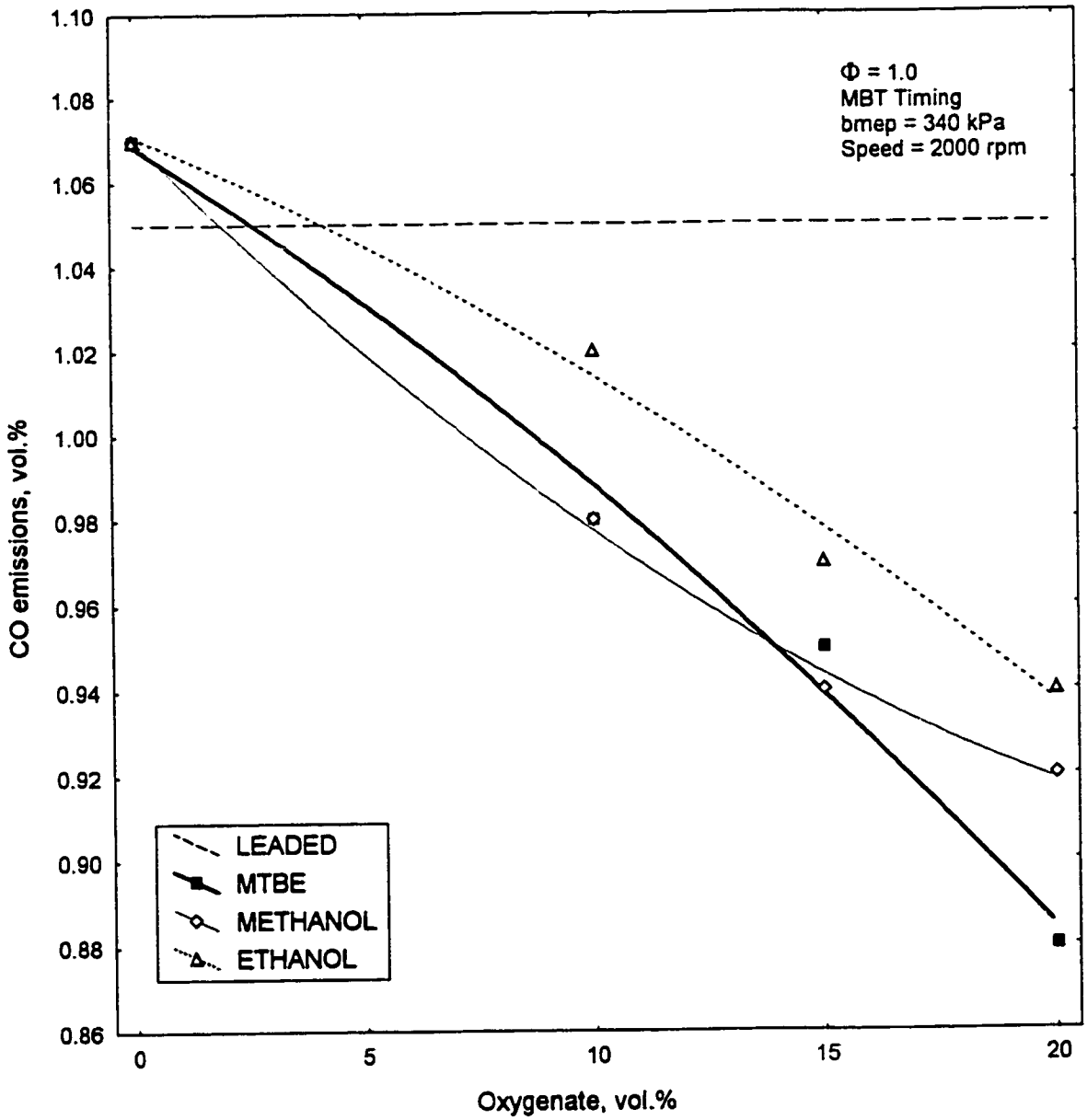

Figure 6-4: Effect of the oxygenates on CO emissions at low load (340 kPa) and mid-range speed (2000 rpm).

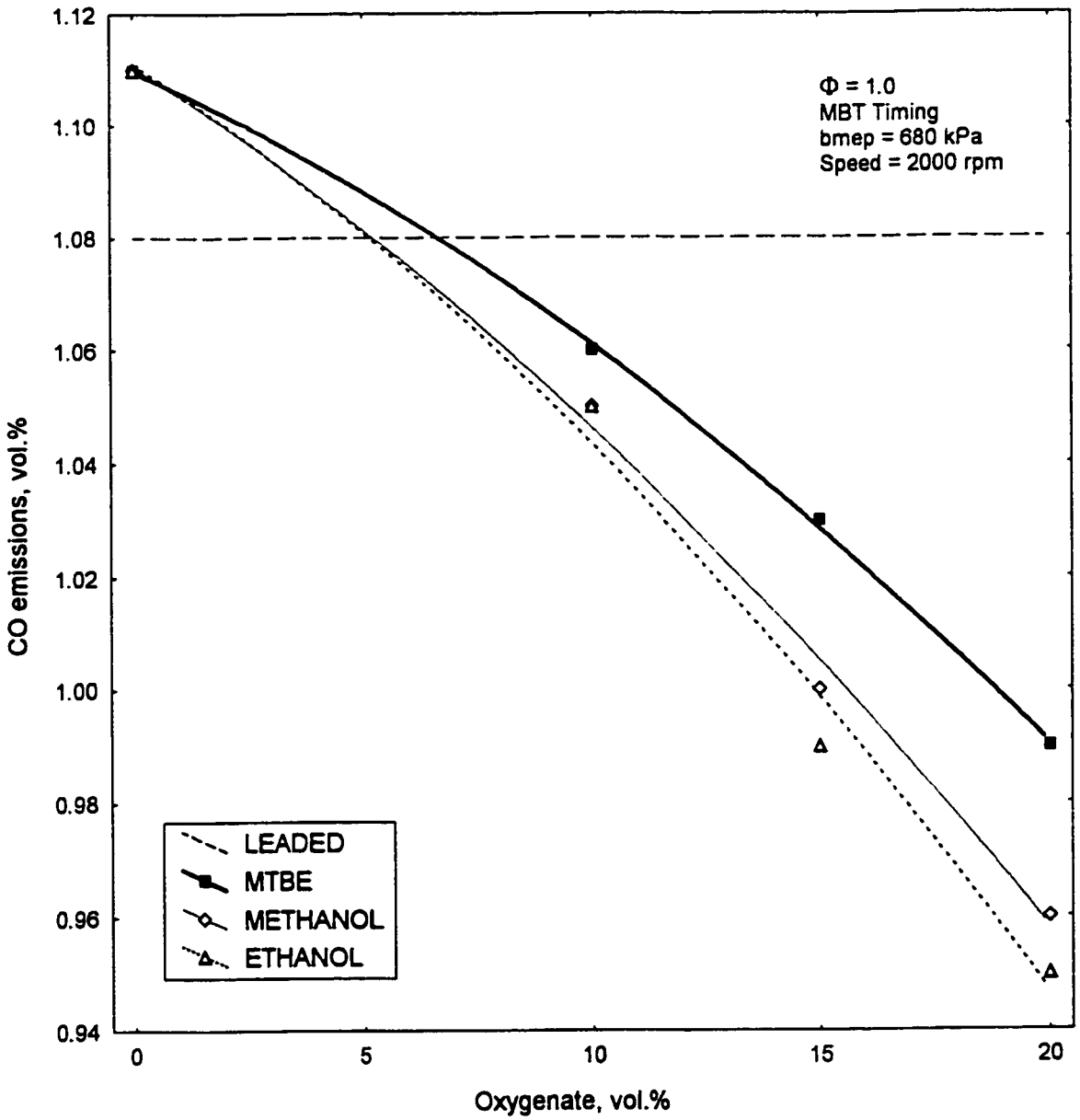

Figure 6-5: Effect of the oxygenates on CO emissions at high load (680 kPa) and mid-range speed (2000 rpm).

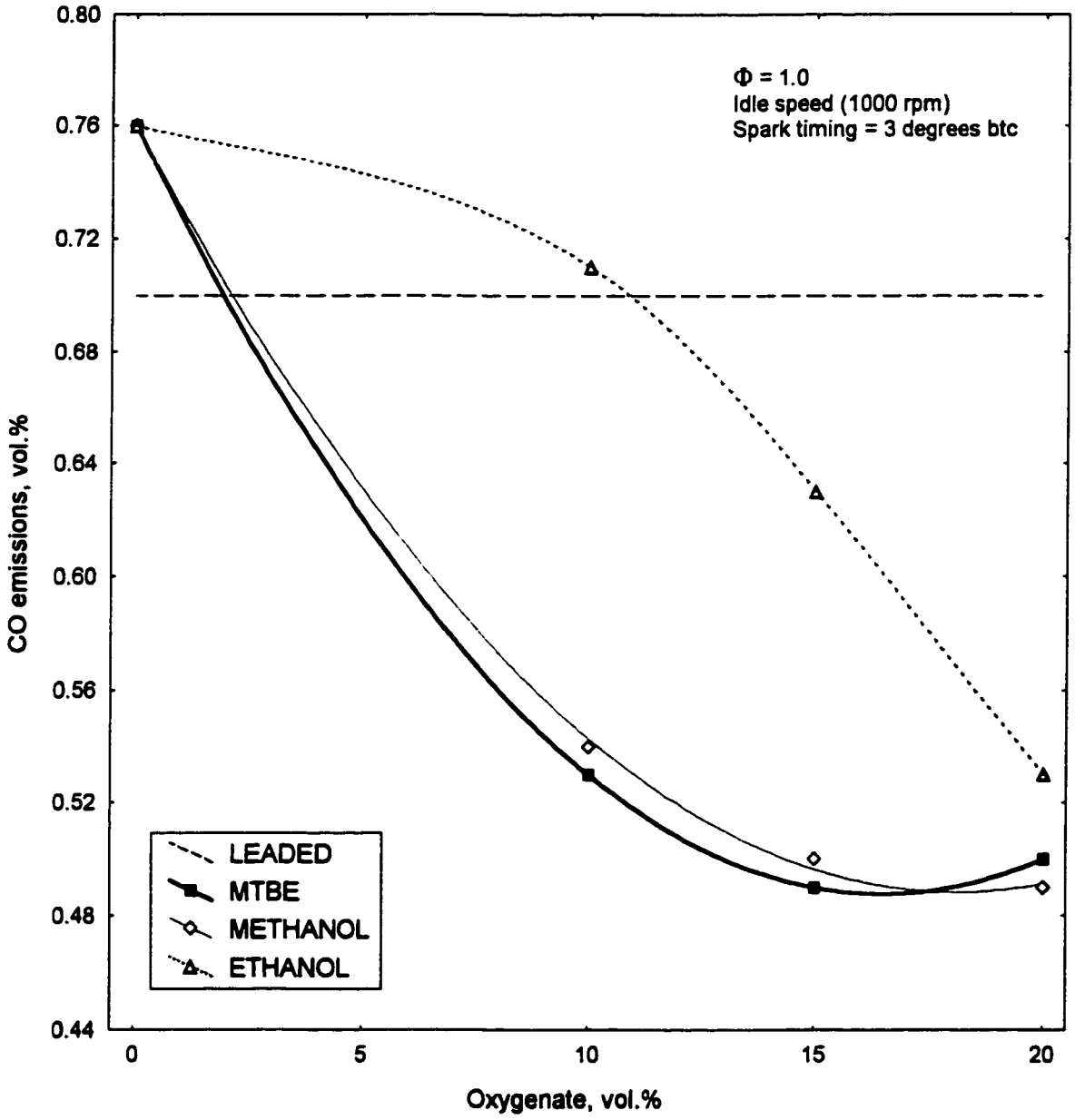

Figure 6-6: Effect of the oxygenates on CO emissions at idle speed (1000 rpm) with spark timing at 3 degrees btc.

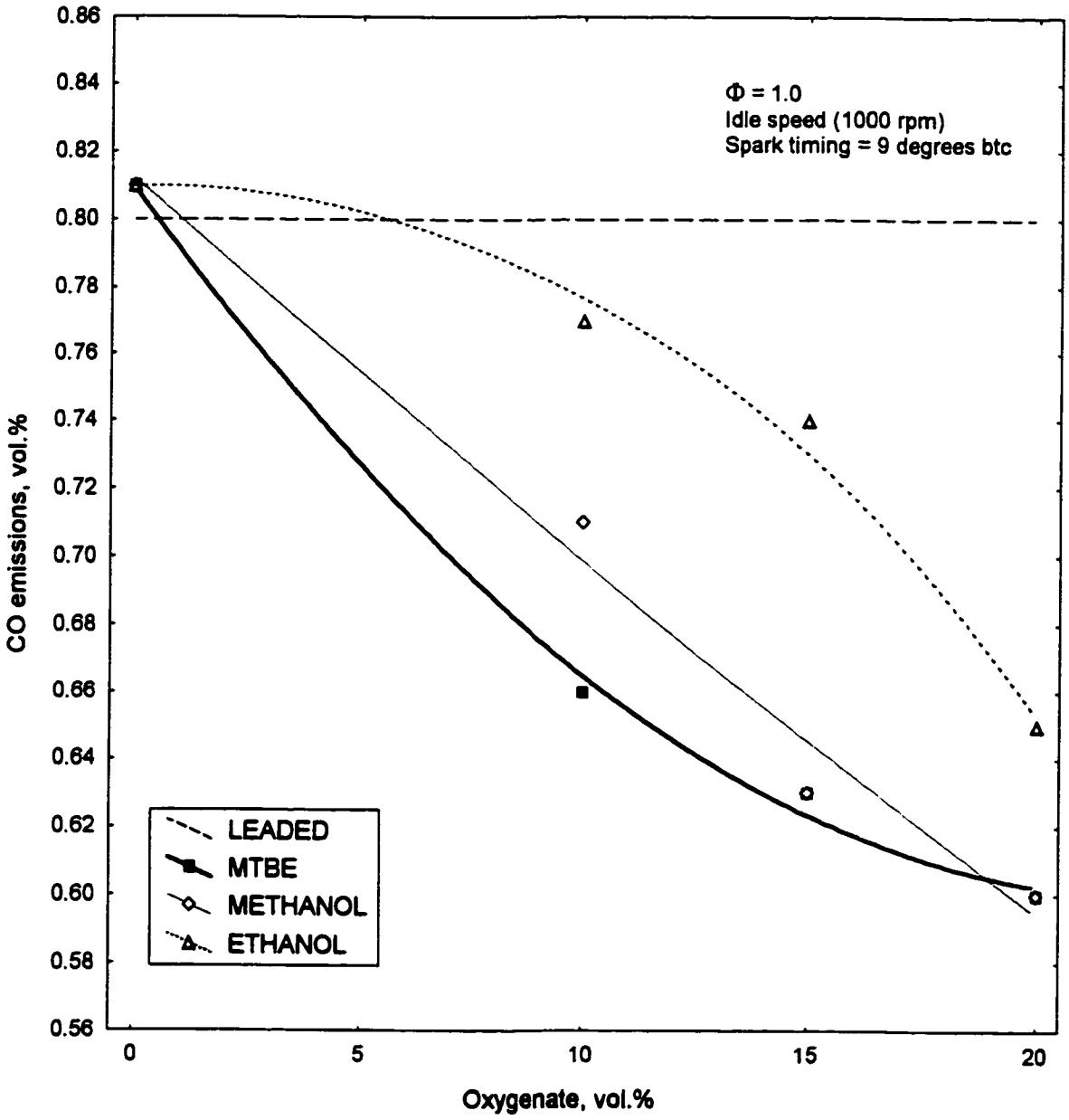

Figure 6-7: Effect of the oxygenates on CO emissions at idle speed (1000 rpm) with spark timing at 9 degrees btc.

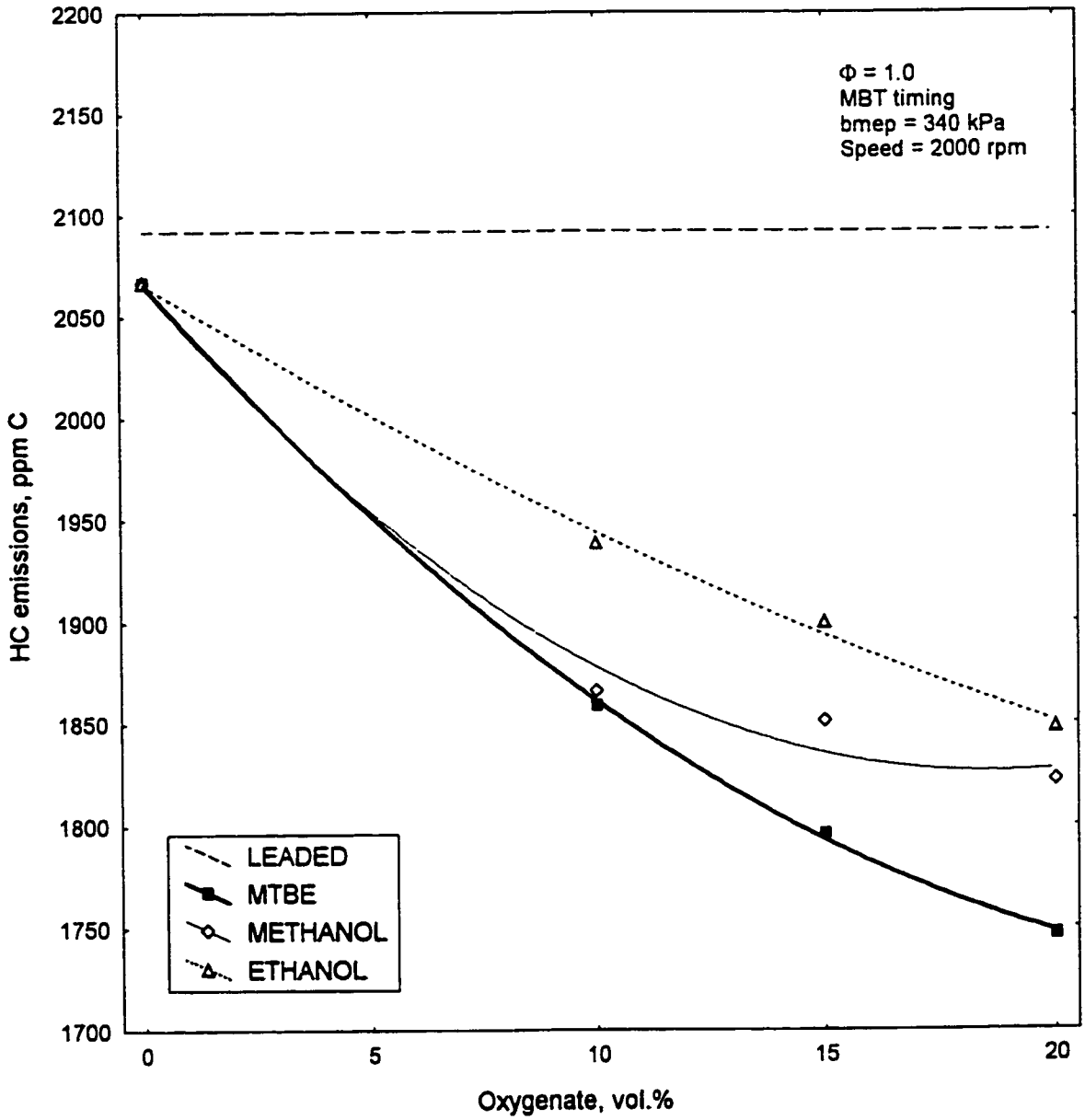

Figure 6-8: Effect of the oxygenates on HC emissions at mid-range speed (2000 rpm) and low load (340 kPa).

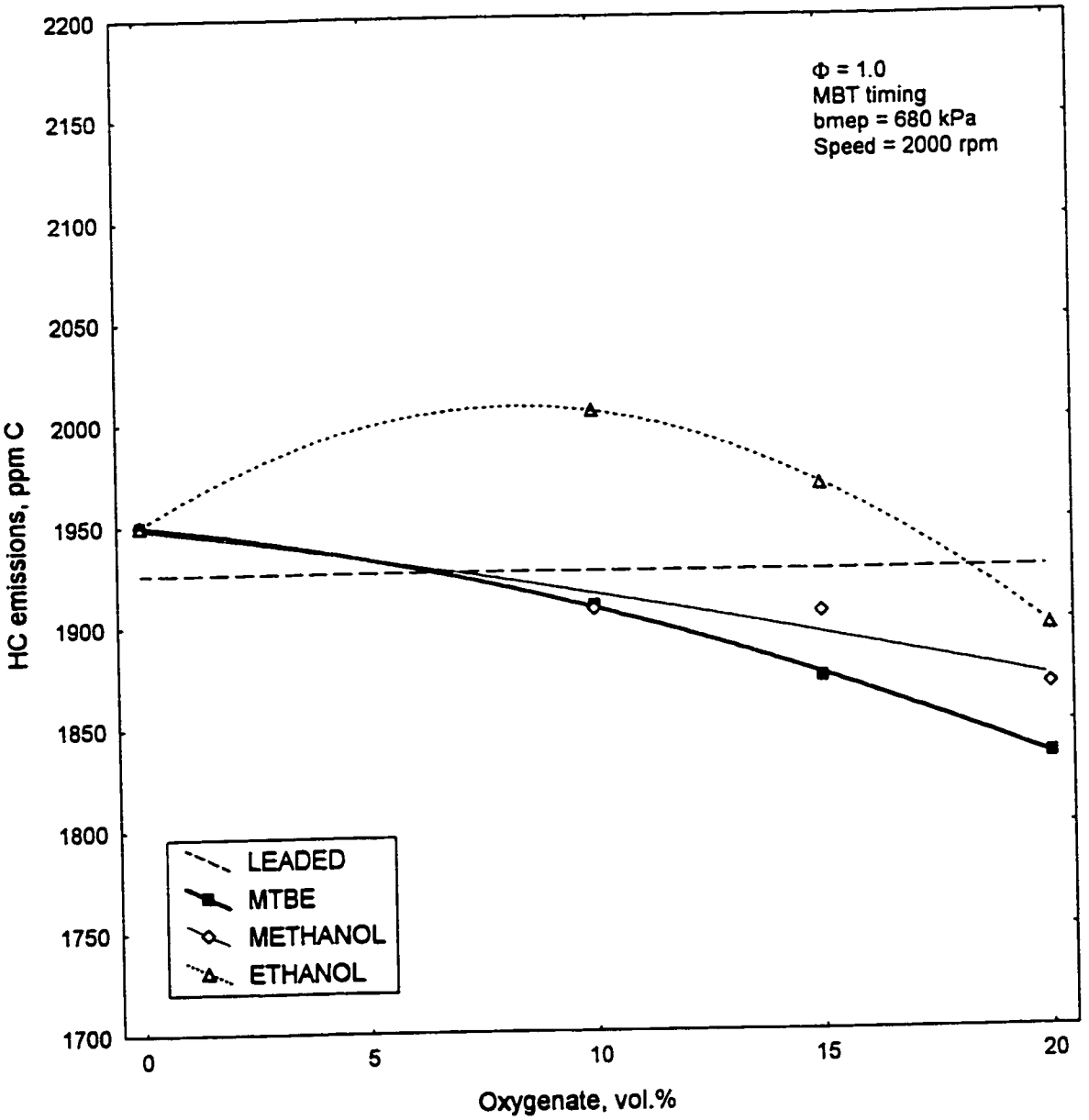

Figure 6-9: Effect of the oxygenates on HC emissions at mid-range speed (2000 rpm) and high load (680 kPa).

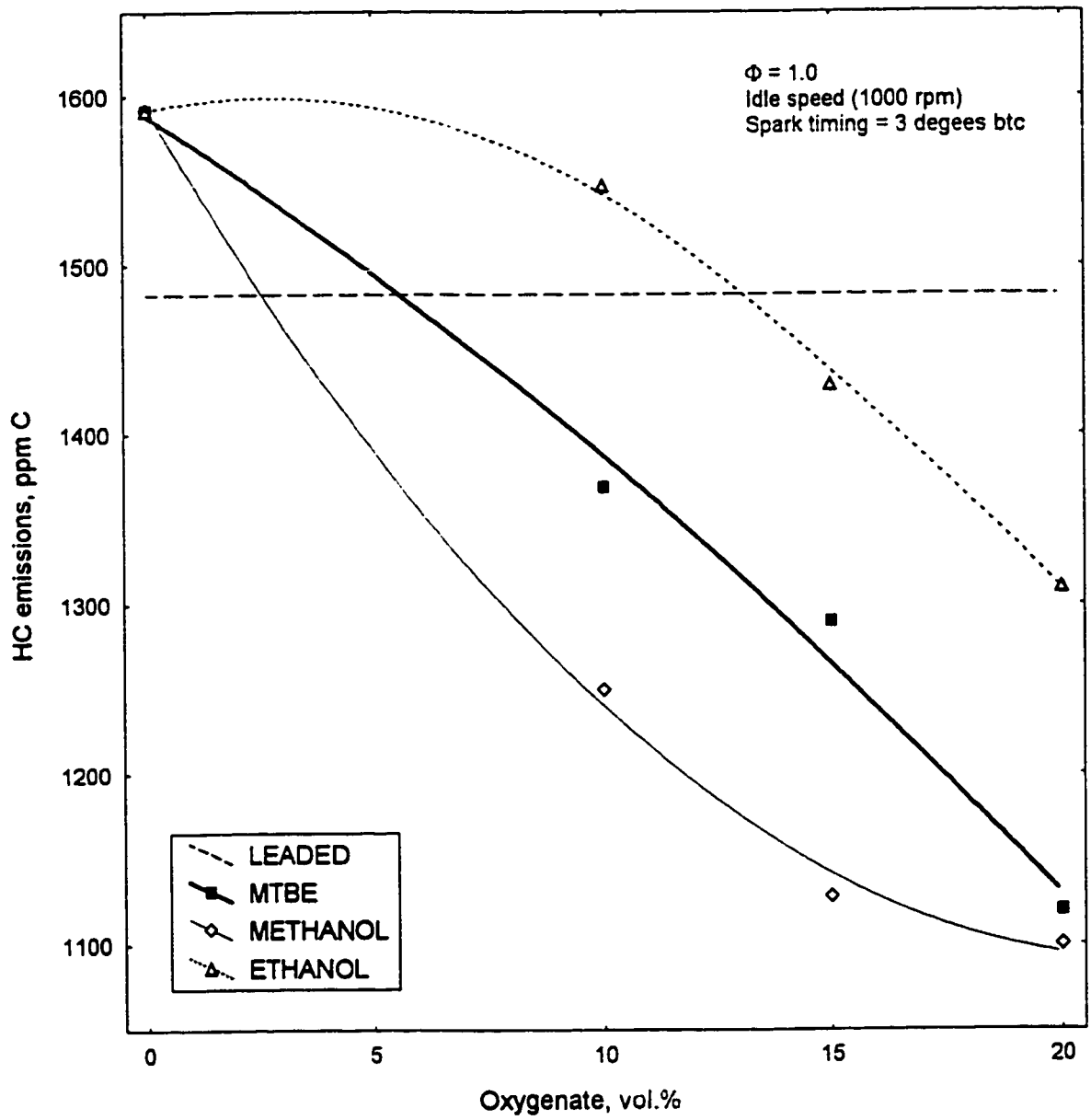

Figure 6-10: Effect of the oxygenates on HC emissions at idle speed (1000 rpm) with spark timing at 3 degrees btc.

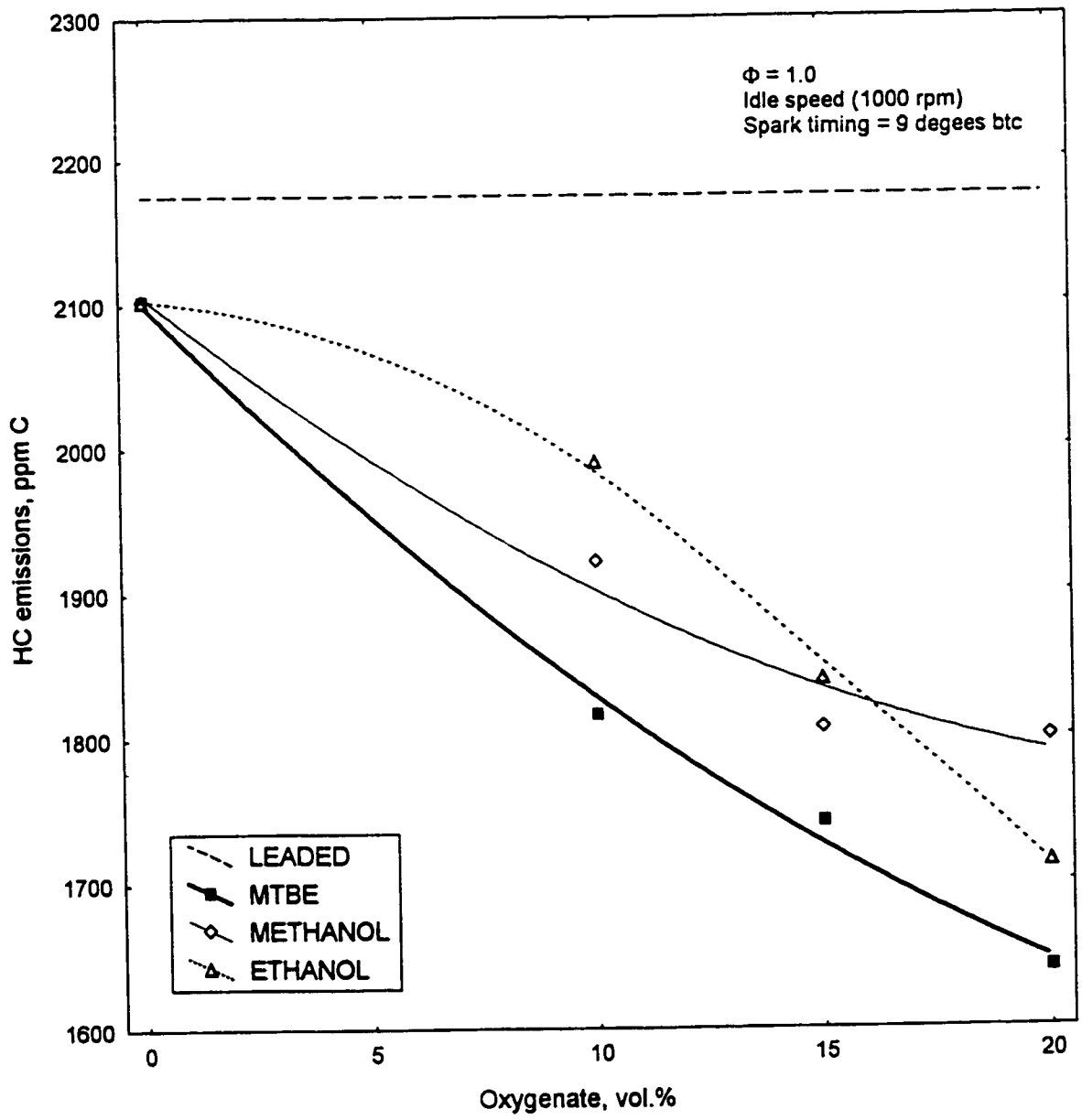

Figure 6-11: Effect of the oxygenates on HC emissions at idle speed (1000 rpm) with spark timing at 9 degrees btc.

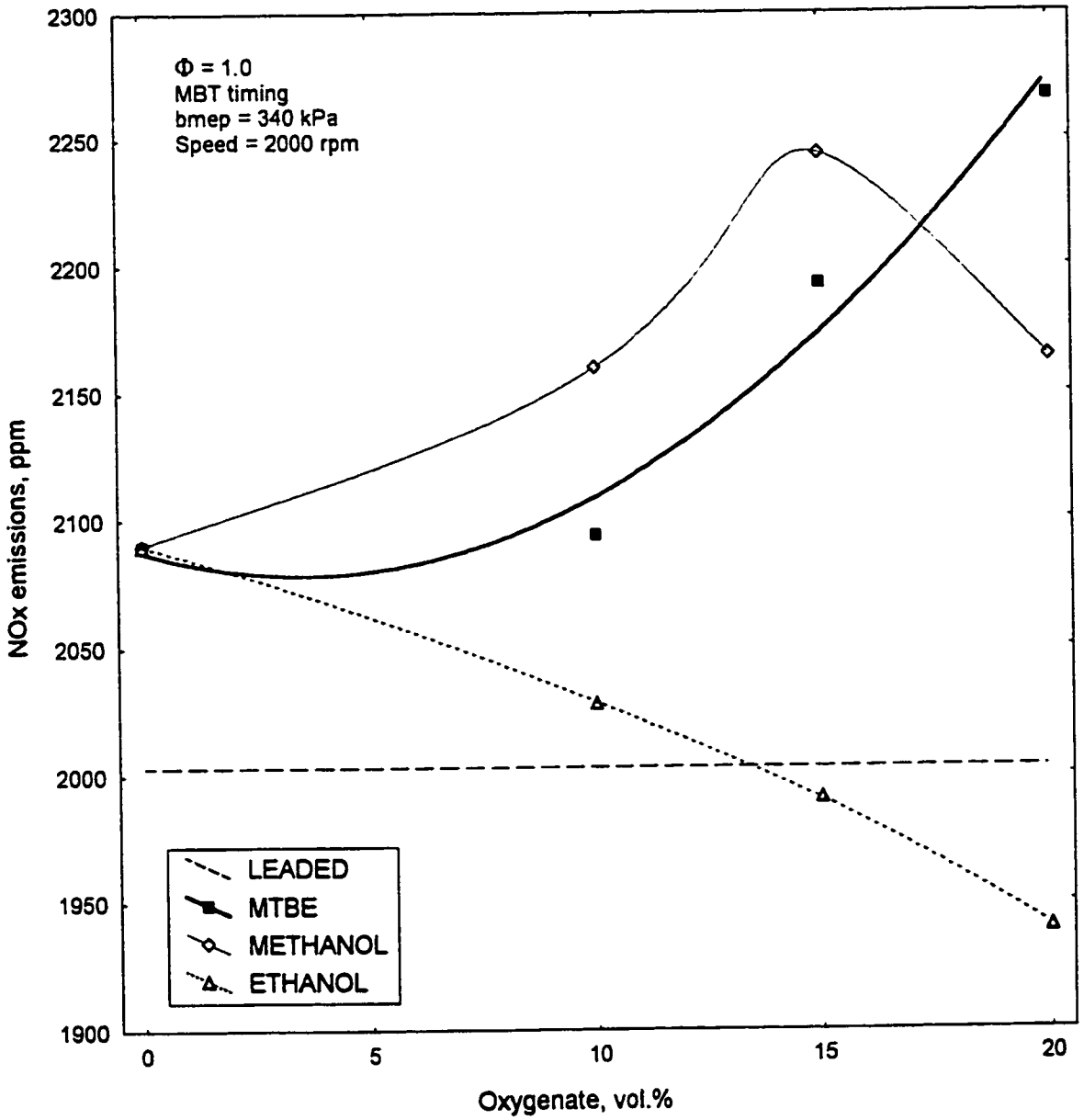

Figure 6-12: Effect of the oxygenates on NOx emissions at mid-range speed (2000 rpm) and low load (340 kPa).

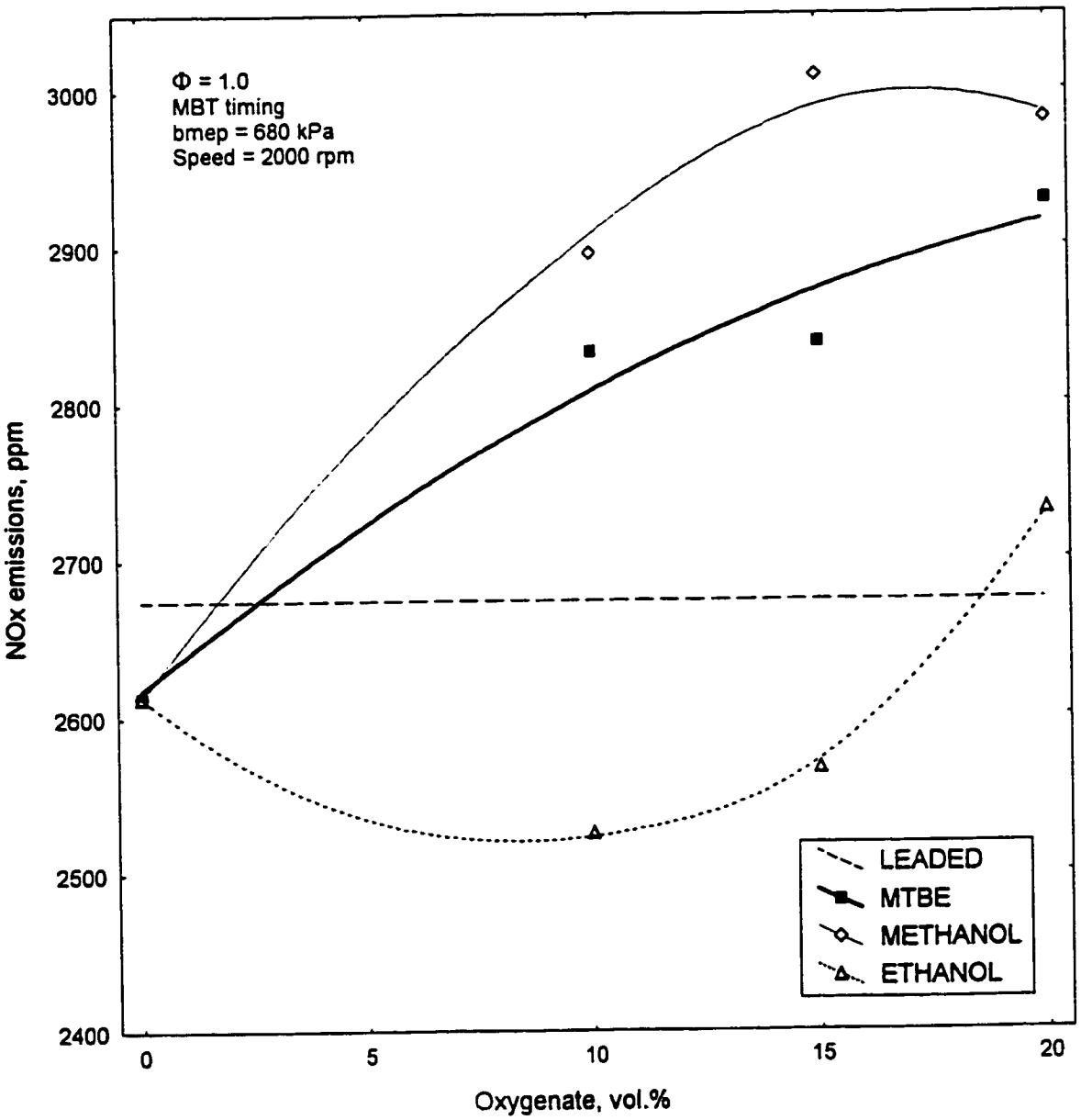

Figure 6-13: Effect of the oxygenates on NOx emissions at mid-range speed (2000 rpm) and high load (680 kPa).

Appendixes

APPENDIX A: EXPERIMENTAL DATA 234
APPENDIX B: MEASUREMENT UNCERTAINTY 246
APPENDIX C: FUEL COMPOSITION 248
APPENDIX D: TEST EQUIPMENT OPERATING PROCEDURE 249

Appendix

A

Experimental Data

Table A-1: Performance data at wide-open throttle, MBT timing, and stoichiometric mixture (Φ=1.0).

(MBT: Maximum Brake Torque Spark Timing, SFC: Specific Fuel Consumption)

Fuel	RPM	1000	1500	2000	2500	3000	3500
Base Fuel	MBT (btc)	6	13	18	21	24	25
	Torque (Nm)	199.1	212.9	214.7	221.4	220.6	221.5
	SFC (g/kWh)	291.80	271.82	260.02	257.08	253.00	259.51
	Intake Temp. (°C)	21	22	24	26	28	28
	Intake Press. (kPa)	100.2	100.2	100.2	100.2	100.2	100.2
	Exh. Temp. (°C)	649.4	730.4	761.6	809.9	846.3	880.0
Leaded	MBT (btc)	13	20	28	31	33	35
	Torque (Nm)	208.6	222.3	222.2	229	225.9	229.2
	SFC (g/kWh)	263.93	256.50	252.34	246.79	250.35	258.82
	Intake Temp. (°C)	20	22	23	26	27.5	27
	Intake Press. (kPa)	99.9	99.9	99.9	99.9	100	99.9
	Exh. Temp. (°C)	623.8	702.0	728.6	779.2	814.3	841.2
10% MTBE	MBT (btc)	9	15	23	26	28	31
	Torque (Nm)	201.30	215.37	219.10	226.83	226.19	229.04
	SFC (g/kWh)	285.34	266.10	249.99	247.48	248.44	249.13
	Intake Temp. (°C)	21	24	27	27	28	28
	Intake Press. (kPa)	99.7	99.7	99.7	99.7	99.7	99.7
	Exh. Temp. (°C)	640.2	716.9	744.9	790.1	827.9	854.6
15% MTBE	MBT (btc)	10	16	25	28	32.5	32
	Torque (Nm)	203.02	217.00	219.32	227.60	227.16	229.07
	SFC (g/kWh)	276.46	255.59	249.94	247.14	248.16	248.25
	Intake Temp. (°C)	21	22	25	27	28	29
	Intake Press. (kPa)	99.5	99.5	99.5	99.5	99.5	99.5
	Exh. Temp. (°C)	631.8	714.1	737.4	783.2	817.7	847.3
20% MTBE	MBT (btc)	12	20	28	31	33	35
	Torque (Nm)	203.94	218.42	218.84	226.64	224.47	226.68
	SFC (g/kWh)	269.64	253.37	245.55	245.76	244.98	247.31
	Intake Temp. (°C)	21	23	25	26	27	28
	Intake Press. (kPa)	99.4	99.4	99.4	99.4	99.4	99.4
	Exh. Temp. (°C)	618.0	699.4	733.8	778.0	817.4	843.3

Table A-1: (cont.)

Fuel	RPM	1000	1500	2000	2500	3000	3500
10% Methanol	MBT (btc)	12	18	25	28	31	33
	Torque (Nm)	204.82	218.82	219.88	227.46	227.59	230.30
	SFC (g/kWh)	290.33	265.71	260.95	250.06	260.68	261.78
	Intake Temp. (°C)	23	25	29	30	33	35
	Intake Press. (kPa)	100.9	100.9	100.9	100.9	100.8	100.8
	Exh. Temp. (°C)	640.9	722.5	753.8	804.2	842.2	871.0
15% Methanol	MBT (btc)	13	19	25	29	31	33
	Torque (Nm)	207.21	223.11	222.09	229.57	229.81	231.44
	SFC (g/kWh)	283.35	269.21	257.98	260.14	256.06	259.07
	Intake Temp. (°C)	29	31	31	32	36	37
	Intake Press. (kPa)	100.6	100.6	100.6	100.6	100.6	100.6
	Exh. Temp. (°C)	631.5	712.0	742.8	792.5	830.0	858.3
20% Methanol	MBT (btc)	14	20	27	30	32	33
	Torque (Nm)	209.56	223.75	223.14	231.88	231.53	233.18
	SFC (g/kWh)	296.16	271.55	261.92	258.42	258.81	266.16
	Intake Temp. (°C)	32	35	35	39	40	40
	Intake Press. (kPa)	100.6	100.6	100.6	100.6	100.6	100.6
	Exh. Temp. (°C)	622.2	701.4	731.8	780.8	817.7	845.7
10% Ethanol	MBT (btc)	10	17.5	24	26	30	32
	Torque (Nm)	206.12	220.95	221.76	229.96	229.89	231.84
	SFC (g/kWh)	284.39	266.20	259.11	257.49	254.98	262.53
	Intake Temp. (°C)	20	23	24	26	27	28
	Intake Press. (kPa)	99.4	99.4	99.4	99.4	99.7	99.4
	Exh. Temp. (°C)	637.7	715.3	736.1	787.6	821.0	846.3
15% Ethanol	MBT (btc)	13	22	27	31	34	35
	Torque (Nm)	206.61	222.20	222.98	231.87	231.24	233.25
	SFC (g/kWh)	286.41	275.28	265.98	262.58	260.00	264.40
	Intake Temp. (°C)	21	20	21	23	25	26
	Intake Press. (kPa)	99.9	99.9	99.9	99.7	99.7	99.7
	Exh. Temp. (°C)	620.7	694.4	727.1	774.1	812.6	838.1
20% Ethanol	MBT (btc)	14	23	28	32	35	36.5
	Torque (Nm)	206.52	221.22	221.59	229.41	228.02	230.68
	SFC (g/kWh)	291.30	273.83	266.63	262.54	258.53	269.72
	Intake Temp. (°C)	24	24	25	27	30	29
	Intake Press. (kPa)	100.2	100.2	100.2	100.2	100.2	100.2
	Exh. Temp. (°C)	620.3	689.2	725.2	773.3	811.8	836.8

Table A-2: Speed effects on exhaust emissions at a constant load of 80 Nm (bmep = 340 kPa), MBT timing, and stoichiometric mixture (Φ=1.0).

Fuel	RPM	1000	1500	2000	2500	3000
Base Fuel	MBT (btc)	20	26	31	34	36
	HC (ppmC)	2387	2724	2067	2128	1893
	NOx (ppm)	1389	1884	2090	2328	2229
	CO (%)	1.02	1.09	1.07	1.13	1.14
Leaded	MBT (btc)	21	27	33	35	38
	HC (ppmC)	2518	2800	2092	2180	1924
	NOx (ppm)	1387	1843	2003	2250	2203
	CO (%)	0.97	1.04	1.05	1.08	1.12
10% MTBE	MBT (btc)	24	27	32	35	38
	HC (ppmC)	2498	2636	1859	2002	1722
	NOx (ppm)	1742	1940	2094	2365	2258
	CO (%)	0.91	1.00	0.98	1.06	1.10
15% MTBE	MBT (btc)	25	28	33	36	39
	HC (ppmC)	2420	2513	1796	1899	1663
	NOx (ppm)	1780	2000	2193	2415	2272
	CO (%)	0.84	0.95	0.95	0.99	1.04
20% MTBE	MBT (btc)	26	29	34	37	40
	HC (ppmC)	2313	2491	1747	1841	1606
	NOx (ppm)	1973	2080	2267	2550	2479
	CO (%)	0.78	0.90	0.88	0.90	0.96
10% Methanol	MBT (btc)	22	27	31	34	37
	HC (ppmC)	2465	2685	1866	1994	1665
	NOx (ppm)	1669	2000	2160	2400	2386
	CO (%)	0.90	0.99	0.98	1.07	1.11
15% Methanol	MBT (btc)	23	27	32	34	37
	HC (ppmC)	2390	2553	1851	1913	1682
	NOx (ppm)	1689	2107	2244	2511	2430
	CO (%)	0.88	0.96	0.94	0.99	1.05
20% Methanol	MBT (btc)	24	28	33	35	38
	HC (ppmC)	2352	2503	1822	1857	1606
	NOx (ppm)	1641	2009	2164	2362	2309
	CO (%)	0.85	0.92	0.92	0.98	1.00
10% Ethanol	MBT (btc)	22	27	31	35	37
	HC (ppmC)	2452	2635	1939	2032	1845
	NOx (ppm)	1413	1751	2028	2295	2175
	CO (%)	0.97	1.02	1.02	1.07	1.10
15% Ethanol	MBT (btc)	24	29	32	36	37
	HC (ppmC)	2394	2501	1900	1959	1743
	NOx (ppm)	1484	1745	1991	2184	2145
	CO (%)	0.93	0.99	0.97	1.01	1.05
20% Ethanol	MBT (btc)	26	30	33	36	38
	HC (ppmC)	2412	2474	1848	1823	1660
	NOx (ppm)	1602	1774	1940	2119	2135
	CO (%)	0.90	0.96	0.94	0.97	1.01

Table A-3: Speed effects on exhaust emissions at a constant load of 120 Nm (bmep = 510 kPa), MBT timing, and stoichiometric mixture (Φ=1.0).

Fuel	RPM	1000	1500	2000	2500	3000
Base Fuel	MBT (btc)	19	25	29	32	34
	HC (ppmC)	2434	2581	1991	2010	1834
	NOx (ppm)	1898	2289	2440	2565	2590
	CO (%)	1.04	1.12	1.13	1.16	1.17
Leaded	MBT (btc)	21	27	31	34	36
	HC (ppmC)	2460	2600	2002	2020	1841
	NOx (ppm)	1910	2250	2371	2505	2531
	CO (%)	1.02	1.11	1.10	1.18	1.18
10% MTBE	MBT (btc)	22	26	30	33	36
	HC (ppmC)	2357	2460	1859	1895	1728
	NOx (ppm)	2277	2506	2607	2768	2784
	CO (%)	0.97	1.04	1.06	1.07	1.12
15% MTBE	MBT (btc)	23	27	32	34	38
	HC (ppmC)	2340	2341	1843	1804	1713
	NOx (ppm)	2468	2600	2669	2830	2844
	CO (%)	0.92	1.02	0.99	1.03	1.09
20% MTBE	MBT (btc)	24	28	33	35	39
	HC (ppmC)	2209	2266	1753	1701	1615
	NOx (ppm)	2553	2649	2710	2910	3067
	CO (%)	0.86	0.96	0.93	1.00	1.06
10% Methanol	MBT (btc)	19	27	29	32	35
	HC (ppmC)	2356	2500	1866	1876	1743
	NOx (ppm)	2167	2573	2651	2712	2717
	CO (%)	0.97	1.04	1.02	1.06	1.11
15% Methanol	MBT (btc)	21	28	30	33	36
	HC (ppmC)	2302	2448	1836	1793	1717
	NOx (ppm)	2355	2631	2689	2780	2872
	CO (%)	0.94	0.98	0.99	1.05	1.07
20% Methanol	MBT (btc)	22	29	31	34	37
	HC (ppmC)	2310	2355	1828	1786	1704
	NOx (ppm)	2267	2576	2681	2725	2749
	CO (%)	0.91	0.96	0.96	1.00	1.03
10% Ethanol	MBT (btc)	19	27	30	33	35
	HC (ppmC)	2467	2548	1980	1978	1814
	NOx (ppm)	1765	2202	2373	2512	2553
	CO (%)	1.01	1.08	1.05	1.08	1.11
15% Ethanol	MBT (btc)	21	28	31	34	35
	HC (ppmC)	2299	2497	1949	1905	1729
	NOx (ppm)	1808	2206	2459	2583	2550
	CO (%)	0.98	1.02	1.00	1.07	1.08
20% Ethanol	MBT (btc)	23	30	32	35	36
	HC (ppmC)	2232	2441	1868	1803	1705
	NOx (ppm)	1903	2267	2545	2638	2546
	CO (%)	0.92	0.98	0.95	1.00	1.05

Table A-4: Speed effects on exhaust emissions at a constant load of 160 Nm (bmep = 680 kPa), MBT timing, and stoichiometric mixture (Φ=1.0).

Fuel	RPM	1000	1500	2000	2500	3000
Base Fuel	MBT (btc)	12	21	27	31	33
	HC (ppmC)	2300	2377	1950	1789	1651
	NOx (ppm)	2156	2383	2613	2763	2858
	CO (%)	1.03	1.09	1.11	1.12	1.18
	Exh. Temp. (°C)	591.3	662.1	699.8	746.7	785.0
Leaded	MBT (btc)	17	27	30	35	37
	HC (ppmC)	2365	2430	1926	1825	1727
	NOx (ppm)	2275	2445	2674	2779	2902
	CO (%)	1.03	1.09	1.08	1.13	1.15
	Exh. Temp. (°C)	569.5	644.3	690.3	740.3	782.9
10% MTBE	MBT (btc)	14	24	29	31	33
	HC (ppmC)	2284	2305	1909	1761	1644
	NOx (ppm)	2406	2574	2833	2852	2876
	CO (%)	1.01	1.06	1.06	1.11	1.14
	Exh. Temp. (°C)	577.7	652.0	698.3	748.0	791.2
15% MTBE	MBT (btc)	16	26	30	32	35
	HC (ppmC)	2197	2172	1873	1735	1555
	NOx (ppm)	2570	2708	2839	2968	2916
	CO (%)	0.97	1.04	1.03	1.07	1.12
	Exh. Temp. (°C)	576.0	649.0	695.0	745.0	788.0
20% MTBE	MBT (btc)	17	27	31	33	36
	HC (ppmC)	2083	2143	1834	1663	1469
	NOx (ppm)	2686	2800	2929	3050	3096
	CO (%)	0.95	0.99	0.99	1.02	1.04
	Exh. Temp. (°C)	573.0	647.0	692.0	742.0	785.1
10% Methanol	MBT (btc)	16	24	28	31	35
	HC (ppmC)	2223	2260	1907	1796	1591
	NOx (ppm)	2571	2808	2896	2984	3053
	CO (%)	0.99	1.03	1.05	1.10	1.10
	Exh. Temp. (°C)	607.8	658.0	699.0	745.0	783.0
15% Methanol	MBT (btc)	18	25	29	32	35
	HC (ppmC)	2200	2206	1905	1746	1590
	NOx (ppm)	2672	2994	3010	3086	3068
	CO (%)	0.95	0.99	1.00	1.04	1.07
	Exh. Temp. (°C)	597.3	654.0	699.2	743.0	781.5
20% Methanol	MBT (btc)	18	25	30	33	36
	HC (ppmC)	2120	2198	1868	1725	1578
	NOx (ppm)	2453	2740	2981	3042	2968
	CO (%)	0.92	0.97	0.96	1.00	1.04
	Exh. Temp. (°C)	585.0	650.0	695.8	741.0	781.4
10% Ethanol	MBT (btc)	13	24	29	32	35
	HC (ppmC)	2326	2489	2005	1832	1756
	NOx (ppm)	2111	2385	2526	2666	2719
	CO (%)	0.98	1.03	1.05	1.07	1.13
	Exh. Temp. (°C)	589.8	650.8	696.3	746.1	782.8

Table A-4: (cont.)

Fuel	RPM	1000	1500	2000	2500	3000
15% Ethanol	MBT (btc)	18	26	30	32	35
	HC (ppmC)	2360	2420	1968	1821	1703
	NOx (ppm)	2115	2310	2567	2754	2746
	CO (%)	0.94	1.00	0.99	1.04	1.06
	Exh. Temp. (°C)	573.1	650.8	695.9	744.3	783.3
20% Ethanol	MBT (btc)	20	28	31	34	36
	HC (ppmC)	2402	2410	1897	1732	1675
	NOx (ppm)	2222	2530	2731	2839	2838
	CO (%)	0.90	0.95	0.95	0.98	1.01
	Exh. Temp. (°C)	567.2	641.0	691.8	740.4	783.8

Table A-5: Speed effects on exhaust emissions at a constant load of 200 Nm (bmep = 850 kPa), MBT timing, and stoichiometric mixture (Φ=1.0).

Fuel	RPM	1000	1500	2000	2500	3000
Base Fuel	MBT (btc)	7	14	20	25	29
	HC (ppmC)	3488	2247	1868	1630	1580
	NOx (ppm)	2359	2452	2510	2615	2650
	CO (%)	1.02	1.01	1.02	1.09	1.13
Leaded	MBT (btc)	12	21	27	30	31
	HC (ppmC)	3347	2350	1960	1700	1625
	NOx (ppm)	2287	2460	2631	2650	2666
	CO (%)	1.01	1.03	1.03	1.11	1.15
10% MTBE	MBT (btc)	9	15	23	28	30
	HC (ppmC)	3856	2398	1937	1641	1586
	NOx (ppm)	2651	2670	2863	2964	2956
	CO (%)	0.98	0.99	0.97	1.08	1.11
15% MTBE	MBT (btc)	11	17	26	29	32
	HC (ppmC)	3566	2314	1914	1591	1554
	NOx (ppm)	2720	2810	2899	3099	3090
	CO (%)	0.94	0.97	0.96	1.04	1.10
20% MTBE	MBT (btc)	12	21	28	30	34
	HC (ppmC)	3479	2291	1831	1568	1383
	NOx (ppm)	2806	2980	3063	3120	3185
	CO (%)	0.92	0.94	0.93	0.99	1.04
10% Methanol	MBT (btc)	10	18	26	30	31
	HC (ppmC)	3371	2414	1991	1724	1539
	NOx (ppm)	2569	2774	3030	3010	3073
	CO (%)	0.98	1.01	1.02	1.08	1.10
15% Methanol	MBT (btc)	13	21	27	29	32
	HC (ppmC)	2815	2231	1976	1716	1553
	NOx (ppm)	2866	3042	3159	3165	3158
	CO (%)	0.94	0.98	0.99	1.03	1.06
20% Methanol	MBT (btc)	15	23	28	31	33
	HC (ppmC)	2500	2198	1881	1655	1502
	NOx (ppm)	2840	3034	3113	3104	3060
	CO (%)	0.91	0.96	0.95	0.99	1.02
10% Ethanol	MBT (btc)	10	18	25	29	31
	HC (ppmC)	3991	2693	2048	1745	1672
	NOx (ppm)	2146	2331	2472	2592	2644
	CO (%)	0.98	1.01	1.02	1.07	1.11
15% Ethanol	MBT (btc)	13.5	23	28	31	33
	HC (ppmC)	3704	2751	2037	1860	1672
	NOx (ppm)	2308	2571	2617	2762	2765
	CO (%)	0.94	0.98	0.97	1.02	1.05
20% Ethanol	MBT (btc)	15.5	25	30	33	35
	HC (ppmC)	2931	2524	1993	1763	1585
	NOx (ppm)	2507	2820	2889	2984	2956
	CO (%)	0.90	0.93	0.93	0.95	1.01

Table A-6: Exhaust emissions versus spark timing at idle speed (1000 rpm) and stoichiometric mixture (Φ=1.0).

Fuel	Timing (btc)	3	6	9	12	15
Base Fuel	HC (ppmC)	1592	1783	2103	2280	2566
	NOx (ppm)	98	89	85	79	78
	CO (%)	0.76	0.80	0.81	0.85	0.98
	Exh. Temp. (°C)	500	470	440	422	408
Leaded	HC (ppmC)	1483	1882	2175	2423	2747
	NOx (ppm)	98	90	86	84	84
	CO (%)	0.70	0.73	0.80	0.87	0.95
	Exh. Temp. (°C)	510	472	437	417	402
10% MTBE	HC (ppmC)	1369	1633	1817	2315	2417
	NOx (ppm)	100	93	88	83	82
	CO (%)	0.53	0.60	0.66	0.82	0.97
15% MTBE	HC (ppmC)	1290	1502	1742	2107	2241
	NOx (ppm)	100	94	89	85	85
	CO (%)	0.49	0.53	0.63	0.71	0.82
20% MTBE	HC (ppmC)	1120	1447	1641	1817	2117
	NOx (ppm)	102	96	91	87	87
	CO (%)	0.50	0.51	0.60	0.70	0.73
	Exh. Temp. (°C)	488	466	449	432	418
10% Methanol	HC (ppmC)	1250	1612	1922	2177	2321
	NOx (ppm)	98	89	83	79	78
	CO (%)	0.54	0.62	0.71	0.86	0.97
15% Methanol	HC (ppmC)	1128	1613	1807	2082	2315
	NOx (ppm)	91	87	81	77	78
	CO (%)	0.50	0.61	0.63	0.71	0.82
20% Methanol	HC (ppmC)	1103	1511	1808	2007	2201
	NOx (ppm)	89	85	80	76	76
	CO (%)	0.49	0.58	0.60	0.66	0.75
	Exh. Temp. (°C)	482	456	434	416	402
10% Ethanol	HC (ppmC)	1547	1729	1990	2317	2402
	NOx (ppm)	94	86	82	78	76
	CO (%)	0.71	0.76	0.77	0.87	0.96
15% Ethanol	HC (ppmC)	1429	1719	1840	2119	2267
	NOx (ppm)	91	84	82	79	76
	CO (%)	0.63	0.70	0.74	0.81	0.93
20% Ethanol	HC (ppmC)	1310	1610	1713	1929	2150
	NOx (ppm)	92	87	82	81	78
	CO (%)	0.53	0.57	0.65	0.72	0.84
	Exh. Temp. (°C)	492	470	445	427	412

Table A-7: Effects of spark timing on exhaust emissions at a constant speed of 2000 rpm, a constant load of 160 Nm (bmep = 680 kPa), and stoichiometric mixture (Φ=1.0).

Fuel	Timing (btc)	10	15	20	25	30	35
Base Fuel	HC (ppmC)	1753	1866	1887	1927	1977	1927
	NOx (ppm)	1728	1934	2194	2416	2721	2908
	CO (%)	0.85	0.93	0.97	1.04	1.08	1.10
Leaded	HC (ppmC)	1830	1919	1932	1960	1992	2011
	NOx (ppm)	1632	1923	2188	2440	2674	2788
	CO (%)	0.85	0.91	1.00	1.04	1.08	1.13
10% MTBE	HC (ppmC)	1588	1750	1795	1870	1937	1915
	NOx (ppm)	1704	1950	2192	2530	2847	3005
	CO (%)	0.82	0.89	0.95	1.01	1.06	1.07
15% MTBE	HC (ppmC)	1527	1621	1708	1850	1873	1888
	NOx (ppm)	1746	1992	2258	2524	2839	3075
	CO (%)	0.81	0.85	0.92	0.96	1.03	1.04
20% MTBE	HC (ppmC)	1390	1600	1657	1750	1837	1819
	NOx (ppm)	1763	2030	2294	2582	2870	3119
	CO (%)	0.79	0.83	0.89	0.93	0.99	1.03
10% Methanol	HC (ppmC)	1720	1770	1887	1907	1938	1946
	NOx (ppm)	1790	2115	2436	2720	2997	3158
	CO (%)	0.78	0.88	0.95	1.00	1.06	1.12
15% Methanol	HC (ppmC)	1606	1769	1826	1847	1916	1897
	NOx (ppm)	1887	2145	2399	2690	2994	3169
	CO (%)	0.76	0.83	0.88	1.00	1.01	1.07
20% Methanol	HC (ppmC)	1601	1710	1819	1845	1868	1874
	NOx (ppm)	1715	2010	2335	2642	2981	3176
	CO (%)	0.74	0.82	0.86	0.91	0.97	1.02
10% Ethanol	HC (ppmC)	1733	1817	1898	1949	1997	2020
	NOx (ppm)	1527	1742	2027	2287	2528	2704
	CO (%)	0.84	0.90	0.95	1.01	1.05	1.06
15% Ethanol	HC (ppmC)	1712	1794	1874	1905	1968	1964
	NOx (ppm)	1609	1791	2108	2347	2571	2776
	CO (%)	0.81	0.85	0.88	0.96	0.99	1.01
20% Ethanol	HC (ppmC)	1652	1786	1854	1866	1889	1893
	NOx (ppm)	1505	1777	2104	2435	2657	2901
	CO (%)	0.71	0.76	0.80	0.86	0.94	0.96

Table A-8: Effects of equivalence ratio on exhaust emissions at a constant speed of 2000 rpm, a constant load of 160 Nm (bmep = 680 kPa), and MBT timing.

Fuel	Φ	0.8	0.9	1	1.1	1.2
Base Fuel	Air/fuel Ratio	18.26	16.23	14.61	13.28	12.17
	HC (ppmC)	1292	1415	1923	2274	2709
	NOx (ppm)	3946	4182	2618	1031	391
	CO (%)	0.07	0.08	1.11	4.20	7.09
	MBT (btc)	32	29	28	26	25
	Exh. Temp. (°C)	666.7	696.6	698.2	674.2	649.5
	SFC (g/kWh)	255	259	272	296	332
Leaded	Air/fuel Ratio	18.26	16.23	14.61	13.28	12.17
	HC (ppmC)	1437	1480	1992	2400	2800
	NOx (ppm)	3985	4271	2674	1053	399
	CO (%)	0.07	0.07	1.08	4.15	7.00
	MBT (btc)	37.5	32	30	28	26
	Exh. Temp. (°C)	650.2	682.3	686.7	665.0	643.5
	SFC (g/kWh)	250	252	263	285	316
10% MTBE	Air/fuel Ratio	17.91	15.92	14.33	13.03	11.94
	HC (ppmC)	1437	1388	1909	2315	2722
	NOx (ppm)	3126	3962	2838	1219	455
	CO (%)	0.10	0.10	1.06	3.91	6.71
	MBT (btc)	36	31	29	27	26
	Exh. Temp. (°C)	651.9	688.3	700.3	677.3	654.2
	SFC (g/kWh)	256	260	274	299	333
15% MTBE	Air/fuel Ratio	17.74	15.77	14.19	12.90	11.82
	HC (ppmC)	1473	1424	1873	2249	2702
	NOx (ppm)	2650	4075	2839	1247	486
	CO (%)	0.07	0.07	1.03	3.67	6.53
	MBT (btc)	37	32	30	28	27
	Exh. Temp. (°C)	646.7	683.1	698.7	675.1	651.6
	SFC (g/kWh)	256	260	274	295	328
20% MTBE	Air/fuel Ratio	17.56	15.61	14.05	12.77	11.71
	HC (ppmC)	1449	1334	1834	2190	2593
	NOx (ppm)	2191	4097	2929	1346	507
	CO (%)	0.08	0.07	0.99	3.55	6.37
	MBT (btc)	37.5	33	31	29	28
	Exh. Temp. (°C)	641.5	679.8	694.7	675.9	657.1
	SFC (g/kWh)	256	261	272	291	327
10% Methanol	Air/fuel Ratio	17.21	15.30	13.77	12.52	11.47
	HC (ppmC)	1437	1445	1907	2315	2798
	NOx (ppm)	3690	4389	2896	1326	482
	CO (%)	0.09	0.08	1.05	3.65	6.31
	MBT (btc)	36	31	28	27	26
	Exh. Temp. (°C)	649.7	685.9	700.8	677.7	654.6
	SFC (g/kWh)	264	270	286	309	346

Table A-8: (cont.)

Fuel	Φ	0.8	0.9	1	1.1	1.2
15% Methanol	Air/fuel Ratio	16.69	14.84	13.35	12.14	11.13
	HC (ppmC)	1447	1411	1905	2294	2744
	NOx (ppm)	3946	4309	3010	1383	530
	CO (%)	0.07	0.07	1.00	3.63	6.25
	MBT (btc)	37	32	29	28	27
	Exh. Temp. (°C)	646.9	681.6	699.3	675.5	651.7
	SFC (g/kWh)	267	274	289	314	347
20% Methanol	Air/fuel Ratio	16.17	14.37	12.94	11.76	10.78
	HC (ppmC)	1473	1394	1868	2292	2716
	NOx (ppm)	3630	4393	2981	1267	519
	CO (%)	0.08	0.08	0.96	3.57	6.17
	MBT (btc)	37.5	33	30	29	28
	Exh. Temp. (°C)	642.6	677.2	696.1	671.6	647.2
	SFC (g/kWh)	269	275	292	313	351
10% Ethanol	Air/fuel Ratio	17.43	15.50	13.95	12.68	11.62
	HC (ppmC)	1409	1476	1971	2342	2771
	NOx (ppm)	3620	3996	2526	998	352
	CO (%)	0.08	0.08	1.05	4.06	7.09
	MBT (btc)	34	31	29	27	26
	Exh. Temp. (°C)	655.1	689.2	698.6	673.6	649.4
	SFC (g/kWh)	260	265	280	305	341
15% Ethanol	Air/fuel Ratio	17.03	15.13	13.62	12.38	11.35
	HC (ppmC)	1429	1463	1960	2369	2842
	NOx (ppm)	3485	4080	2570	1037	406
	CO (%)	0.09	0.09	0.99	4.10	6.81
	MBT (btc)	35	32	30	28	27
	Exh. Temp. (°C)	654.8	688.6	694.3	665.4	641.6
	SFC (g/kWh)	262	269	286	312	350
20% Ethanol	Air/fuel Ratio	16.62	14.77	13.30	12.09	11.08
	HC (ppmC)	1490	1442	1878	2354	2832
	NOx (ppm)	3335	4327	2731	1220	461
	CO (%)	0.09	0.09	0.95	3.93	6.50
	MBT (btc)	37	34	31	30	29
	Exh. Temp. (°C)	642.8	678.6	689.5	664.7	639.6
	SFC (g/kWh)	264	273	291	319	358

Appendix

B

Measurement Uncertainty

B.1 Precision Uncertainty

The typical precision uncertainties for the CO, THC, NOx and torque measurements with 95% confidence were found to be as follows:

Measurement	Precision Uncertainty
CO	0.4 %
THC	0.1 %
NOx	0.93 %
Torque	0.09 %

Example of precision uncertainty calculation:

For the METH20 blend at load of 160 Nm (bmep = 680 kPa), speed of 2000 rpm, MBT timing, and $\Phi = 1.2$, the recorded concentration of total hydrocarbons (THC) varied as follows:

Reading No.	THC Concentration, ppmC
1	2726.3
2	2723.7
3	2720.7
4	2721.5
5	2723.6
6	2712.3
7	2711.8
8	2714.8
9	2712
10	2717.1
11	2716.2

Reading No.	THC Concentration, ppmC
12	2719.1
13	2725.1
14	2721
15	2718.6
16	2728.2
17	2727.9
18	2717.5
19	2713.9
20	2711.1
21	2716.9
22	2709.1

The mean concentration therefore is $\overline{C} = 2718.564$ ppmC, and the standard deviation is $S_x = 5.5376$ ppmC. Using the Student's *t*-distribution (which fits small samples), the population mean μ_C lies in the range:

$$\mu_C = \overline{C} \pm t\frac{S_x}{\sqrt{n}}$$

where *t* is the probability distribution, and n is the sample size.

For 95% confidence and a sample size of n = 22, the probability distribution t = 2.074 [ref. 40], therefore:

$$\mu_C = 2718.564 \pm 2.074\frac{5.5376}{\sqrt{22}} = 2718.564 \pm 2.4486\,\text{ppmC}$$

The precision uncertainty in this case with 95% confidence is 2.4486 ppmC, which is about 0.1%.

B.2 Bias Uncertainty

For the CO, THC, and NOx analyzers, the bias uncertainty can be estimated using the following information obtained from the equipment manuals:

	CO Analyzer	THC Analyzer	NOx Analyzer
Range	5000 ppm – 10 vol.%	10 – 50000 ppmC	10 – 10000 ppm
Repeatability (Zero)	< ±1% Full Scale	< ±1% Full Scale	< ±1% Full Scale
Repeatability (Span)	< ±1% Relative Scale	< ±1% Relative Scale	< ±1% Relative Scale
Drift (Zero)	< ±1% Full Scale / 24h	< ±1% Full Scale / 24h	< ±1% Full Scale / 24h
Drift (Span)	< ±1% Full Scale / 24h	< ±1% Full Scale / 24h	< ±1% Full Scale / 24h

Appendix C

Fuel Composition

Table C-1: Test fuels chemical composition, stoichiometric air/fuel ratios, and percent of oxygen weight.

	Oxygenate vol.%	Water In Oxygenate vol.%	C	H	O	$(A/F)_s$	Oxygen wt.%
Gasoline	0	0	8	15	0	14.61	0
MTBE	100	0	4.2301	10.1523	0.8460	11.76	18.18
MTBE10	10	0	5.3976	10.3425	0.0846	14.33	1.77
MTBE15	15	0	5.3327	10.3319	0.1269	14.19	2.66
MTBE20	20	0	5.2678	10.3214	0.1692	14.05	3.55
Ethanol (crude)	100	0.06294	3.2284	10.3831	1.9632	8.30	39.00
ETH10	10	0.06294	5.2974	10.3656	0.1963	13.95	4.08
ETH15	15	0.06294	5.1824	10.3666	0.2945	13.62	6.10
ETH20	20	0.06294	5.0675	10.3675	0.3926	13.30	8.11
Methanol	100	0	2.4766	9.9063	2.4766	6.47	50
METH10	10	0	5.2222	10.3179	0.2477	13.77	5.15
METH15	15	0	5.0697	10.2950	0.3715	13.35	7.71
METH20	20	0	4.9171	10.2722	0.4953	12.94	10.27

Appendix

D

Test Equipment Operating Procedure

The following steps describe briefly the general procedure of operating the test equipment. For more detailed information on operation and calibration, you can refer to the equipment manuals.

a. Preparatory Steps:

- To operate the engine with the open-circuit water cooling system, open the valves labeled W2, W3, and W12. Valves labeled W7 to W11 should be always open. Other valves should be closed. The water pressure gauge should read about 20 psig.
- Open the air line valve to operate the pneumatic valves that control the water and oil heat exchangers.
- Turn on the engine ignition switch and the throttle actuator.
- Open the fuel line valve F7 and close valves F6 and F8. The three-way valves F5 and F2 should be directed to the metering unit. Valves F1, F4, and F9 should be always open.
- Turn on all the gas cylinder lines and adjust the pressure to about 1 bar.
- If they are pressed, release the two engine stop switches (one inside the test engine room and the other on the controlling keyboard).

b. Switching on the Cell Controller and the Gas Analyzers:

- Switch on the Cadet-10 unit (cell controller) and the CAS unit. Press on the "reset" button to silence the alarm.
- Start the gas analyzer by switching on the OVN heated analyzer (located inside the engine room), and after the "OPE" light changes to single flash, turn on the main switch on the MEXA unit.
- After the MEXA computer boots up, confirm the date and time then select the standby mode. If the flashing "Pause" does not change to "STBY", press again the "Stand by" button in the stand by menu.
- Allow some time for system warm-up and to alarms to be cleared.

c. Starting the Test Run:

- From the Cadet-10 main program, start up new test by choosing "Start Test" from the "Run" menu.
- Select the specific engine you have and enter the test name. Then press the function key F3 to start the test run.

- Start the engine by pressing the "Crank" button (Blue) while pressing the "Reset" button (green). Release the crank button once the engine starts and release the reset button after the engine oil alarm goes off. Allow some time for engine warm-up.

d. Setting the Test Conditions:

- To set the engine test conditions manually, press "M" on the controlling keyboard to select the manual mode.
- Assign a control mode ("T" for torque, "S" for speed, and "D" for direct) to the keyboard knobs ("Dyno" and "Throttle").
- Adjust the engine speed and load using the keyboard knobs.

e. Measuring Gas Concentrations:

- To measure the exhaust gas concentrations, make sure that the alarms are cleared then press the tailpipe "STBY" button and choose "Measure".

f. Tracing and Exporting the Measured Values:

- To trace the measured values, run the trace program on the CAS computer. Open the file you are working on and select the channels you want to trace (up to 6 channels). Press "Run" to start tracing.
- You can export the traced data to a file by choosing the "Export" function in the "File" menu.

g. Shutting Off the Analyzers:

- To shut off the analyzers, change the measuring mode to stand by mode by pressing "Reset" in the tailpipe menu. Then choose "Pause" from the stand by menu.
- Press on "Menu" button and choose "Utility", then press "shut down". After the system is down, turn off the main power of the MEXA unit.
- Turn off the oven and close the gas valves.

h. Shutting Off the Engine/Dyno System:

- To shut off the engine/dyno system, cancel the loads on the engine and stop the engine by pressing F1 on the controlling keyboard. Press F1 again to close the program.
- Close "Windows" and turn the Cadet-10 unit off. On the CAS unit, close "Windows" and turn off the computer.
- Close the water cooling system and the fuel lines.

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